Clean Coal Diesel Demonstration Project

Final Report

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DISCLAIMER:

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List of Acronyms and Abbreviations

A&E	Architect and Engineer
BMEP	Brake mean effective pressure
BSFC	Brake-specific fuel consumption
BTDC	Before top dead center
BTU	British thermal units
CA	Crank angle
CAA	Clean Air Act
COV	Coefficient of variation
CWF	Coal-water fuel
CWS	Coal-water slurry
DF2	Diesel fuel no. 2
ECS	Emissions control system
EERC	Environmental and Energy Research Center (North Dakota)
FME	Fairbanks Morse Engine
Hp	Horsepower
ID	Induced draft
IMEP	Indicated mean effective pressure
kW	Kilowatt
kWh	Kilowatt hour
LSC	Model "LSC" engine supplied by Cooper-Bessemer
MIBC	Methyl isobutyl carbinol
MMBtu	Million BTU
MPa	Mean pascal (unit of pressure)
MW	Mega watt
NSPS	New Source Performance Standard
O&M	Operation and maintenance
PLC	Programmable logic control
Ppm	Parts per million
PSI	Pounds per square inch (also, Physical Sciences, Inc.)
RPM	Revolutions per minute
SCR	Selective catalytic reduction
SMD	Sauter mean diameter
TDC	Top dead center
THL	True heavy liquid (organic)
tph	Tons per hour
UAF	University of Alaska, Fairbanks
WC	Tungsten

Introduction

Clean-Coal Diesel Technology for Industrial and Municipal Power Plants

A Clean Coal Diesel project was undertaken to demonstrate a new Clean Coal Technology that offers technical, economic and environmental advantages over conventional power generating methods. This innovative technology (developed to the prototype stage in an earlier DOE project completed in 1992) enables utilization of pre-processed clean coal fuel in large-bore, medium-speed, diesel engines. The diesel engines are conventional modern engines in many respects, except they are specially fitted with hardened parts to be compatible with the traces of abrasive ash in the coal-slurry fuel. Industrial and Municipal power generating applications in the 10 to 100 megawatt size range are the target applications. There are hundreds of such reciprocating engine power-plants operating throughout the world today on natural gas and/or heavy fuel oil.

The following target performance specifications of the mature commercial embodiment of the Clean-Coal diesel, if achieved, will make this technology quite competitive:

- 48% efficiency
- $1300/kW installed cost
- Emission levels controlled to 50-70% below New Source Performance Standards

Demonstration Project Synopsis

The Clean-Coal Diesel technology was developed over the 1982-1993 period under a series of DOE-funded projects, culminating in a successful proof-of-concept prototype test based on a modified Cooper-Bessemer six-cylinder 1800 kW diesel engine which was operated for over 1000 hours on clean coal fuel. The planned demonstration which is described in this report was seen as the next logical step to position the coal diesel technology for commercialization. Phases I and II consisted of Design and Construction of a 6200 kW, 18-cylinder single engine Coal-Diesel Facility, leading to Phase III which was to be the demonstration test. The University of Alaska campus in Fairbanks, Alaska, was the primary host site. At this location, the University constructed the Clean Coal Diesel system as a 6.2 MW diesel power plant addition. The UAF plant utilizes local coal brought by truck from Usibelli's mine in Healey, AK. A 5-ton per hour coal-water-fuel processing plant located in Alaska was also planned as part of this demonstration project. However, after the project was well underway, the necessary commercial partners did not step forward to provide needed matching funds (and additional fuel customers) to build the coal-water fuel processing plant and to operate the 18-cylinder engine facility on coal-water fuel. A limited number of commissioning tests were conducted in 2001-2004 using the 18-cylinder engine facility to demonstrate readiness to operate on CWF. (Those tests continued only as long as matching funds from UAF allowed.) Therefore with DOE approval the demonstration tests were relocated in 2003 to the engine manufacturer's test facility in Wisconsin, and plans were put in place to utilize bituminous coal based CWF processed in Pennsylvania. Another unforeseen development was that this Wisconsin test facility became unavailable late in 2004 (after initial engine tests were completed in April 2004) because the facility was part of a facility renovation and restructuring project. Nevertheless a significant number of technical advances were made, and these are summarized in this report for the benefit of future stationary engine engineering teams that undertake the commercialization of systems for coal-diesel power production.

The third Phase of the project was led by TIAX, with Arthur D. Little, Inc. leading Phases I and II. The team consisted of the University of Alaska, Fairbanks (UAF), the Cooper-Bessemer Reciprocating Division of Cooper Energy Services (Cooper), Fairbanks-Morse Engine, CQ, Inc., the Energy and Environmental Research Center (EERC), and the Usibelli Coal Company. The Ohio Coal Development Office (OCDO) provided valuable support during Phase I of the project.

Clean-Coal Diesel Development at Cooper-Bessemer leading to this Demonstration Project

From 1985 to 1993, Cooper-Bessemer conducted extensive research and development work to burn coal-based fuel in a diesel engine, under the sponsorship of Morgantown Energy Technology Center (now part of NETL) of the U.S. Department of Energy, and in cooperation with Arthur D. Little, Inc., and other leading U.S. companies. The research work on a single-cylinder research engine between 1985 and 1988 firmly established the feasibility of burning Coal Water Fuel (CWF), which is a mixture of equal parts by weight of finely powdered coal and water. This led to undertaking a five-year project in 1988 to develop a six-cylinder 1800 kW pre-production version of the diesel engine to operate on CWF. This project which was completed in 1993 yielded very encouraging results:

- Cooper's 1800 kW and 200 kW research engines were operated for over 1200 hours, developing full power with good fuel consumption, without any deposits inside the engine. Fuel injectors performed flawlessly on the very challenging coal slurry fuel.
- New technologies were developed to protect sensitive engine components such as piston rings, valves and turbocharger rotors from the wear due to constituents in the coal.
- A full scale emissions control system demonstrated the ability to reduce NO_x, SO_x, and particulate emissions to levels significantly below current and anticipated future regulatory limits.

In 1994, the Department of Energy awarded Cooper-Bessemer and Arthur D. Little the CCT-V demonstration project which is reported in this document. The objective of the demonstration was to prove that the coal diesel technology is durable and reliable by operating a significant number of hours in a realistic power plant operating setting.

As a result of developments following the withdrawal of the original host site (Easton, Maryland), in 1996 Arthur D. Little reached agreement with DOE to re-site the Clean Coal Diesel Demonstration to Fairbanks, Alaska. The coal diesel facility was installed in a campus energy park in 1998 to serve as a coal-capable addition to the power plant owned and operated by the University of Alaska, Fairbanks.

Background on Coal-Diesel Technology Development leading to this Demonstration

We wish to acknowledge the wealth of work on coal-fueled diesel engines prior to and concurrent with the 1985-1993 Cooper-Bessemer project that provided a foundation for the success of that 1800 kW prototype engine. The coal diesel had been invented by Rudolf Diesel in 1898-1906 and subsequently developed by Pawlikowski in 1916-1928, as chronicled by Soehngen (1). The advances made in the US from 1957-1993 on coal-diesel technology were also quite significant (2,3,4,6). An excellent overview of this early work as well as the coal diesel technology development up to 1983 was published by Caton and Rosegay (5). Sponsored in most part by DOE, several other prominent companies supplying generating equipment for large reciprocating engine power plants initiated development projects focused on coal-diesel technology. These companies included:

- Sulzer low-speed engine tests led by Thermo-Electron (1979-1987); see Nydick, et al (7)
- SwRI bench-top rig and single-cylinder tests of coal-slurry fuel injection and combustion (1957-1993); see Tracy (27) and Ryan et al (8, 9, 10, 11).
- General Electric, Texas A&M, and DDC used modeling of coal ignition and combustion in medium-speed reciprocating engines to guide selection of coal particle size in the slurry, and mixture preparation for satisfactory ignition (12, 13, 14, 15, 16).
- Caterpillar (17, 83, 84) developed a novel coal fuel pre-processor to avoid the challenges of solid particle injection.
- With DOE support, General Electric and Cooper-Bessemer -- Arthur D Little developed the necessary technology for coal-water fuel pumping, storage and injection (18-32). The same teams developed promising hard material solutions for the severe wear observed in the holes of fuel injector tips (33-40).

Contract R&D and engine specialist firms such as SwRI and Ricardo joined these various teams to advance the technology for in-cylinder coal slurry spray mixing, ignition, and combustion (41-60). Researchers at several universities performed supporting research, including Virginia Tech, MIT, University of California, Penn State and Texas A&M. Hundreds of engine operating hours on coal fuel were accumulated, and the following is only a partial list of significant team efforts on coal-fueled engine testing (61-74):

- General Electric coal-fueled engine development culminating in a twelve-cylinder locomotive engine demonstration and two hundred hours of operation; see Hsu and Confer (19, 85-88)
- Cooper-Bessemer coal-fueled diesel development including over 1200 hours of operation on a six cylinder 1800 kW engine (28, 29, 40, 72, 75, 91, 92).

The present demonstration project could not have been undertaken without important advances in formulating coal slurry properties conducive to good spray behavior in engines (75, 76, 77, 78). Finally, the wear of engine parts such as lube oil, piston rings and liners was reduced through discovery of suitable hard coatings (79, 80, 81, 89). Useful overviews of this and other related coal-fueled diesel work by US-led teams in the modern era (1957-1994) have been published by Caton and Rosegay (5) and Ryan (92). Since the overview by Ryan was an integral part of the technology development leading to the present demonstration project, and includes in-depth assessments of the Cooper-Bessemer prototype engine subsystems such as the injector and the wear mechanisms of hard parts, it is particularly relevant as background to the present demonstration. Accordingly, excerpts from this in-depth coal-diesel technology overview are included in Appendix D.

Most observers agree that the technology was transformed from a historical curiosity to a pre-production promising technology for distributed on-site power plants during this era. The

Cooper-Bessemer 1800 kW prototype emerged as the most promising configuration in 1994 and was the platform and motivation for the demonstration project reported here.

Executive Summary

A Clean Coal Diesel project was undertaken to demonstrate a new Clean Coal Technology that offers technical, economic and environmental advantages over conventional power generating methods. Unlike most clean coal technologies, the clean-coal diesel is uniquely positioned to address small-to-medium sized power plant applications in the 10MW to 100 MW capacity range, including cogeneration and on-site industrial power. Such plants are not necessarily base loaded, and the diesel engine is designed to accommodate rapid start up and load changes; in fact the coal diesel can readily switch fuel source on the fly between conventional natural gas or fuel oil to the nominally lower cost coal. In contrast with IGCC and other larger scale fluid bed gasifier technology platforms, the unique approach taken here is to utilize coal as a solid fuel with only minimal (low-cost) processing to micronize and remove ash and sulfur. This "minimalist" coal processing approach is only possible because the diesel engine is inherently robust and tolerates heavy fuels that are not feasible to burn in gas turbines. The objective of the demonstration was to prove that the coal diesel technology is durable and reliable by operating a significant number of hours in a realistic power plant operating setting.

The key accomplishments of the project can be summarized as follows:

- Designed, constructed, and permitted a 6200 kW, 18-cylinder engine Coal-Diesel facility, suitable to be the demonstration test. The University of Alaska campus in Fairbanks, Alaska, was the primary host site. Special engine parts and subsystems were acquired which made the engine coal-fuel ready.

- Tested the source coal (sub-bituminous coal from the Usibelli mine in Healey, AK) for compatibility with the engine and novel fuel injector system.

- Designed a 5-ton per hour coal-water-fuel processing plant for upgrading sub-bituminous coal using the novel hydrothermal process developed in part by UNDERC.

- Separately designed and operated a coal-water fuel processing plant suitable for Appalachian bituminous coals. Tests on several bituminous coals showed these source coals were also compatible with good coal-diesel engine operation.

- Developed a prototype Fairbanks Morse coal-diesel engine, including novel injectors equipped to avoid erosion and/or clogging by the abrasive coal-slurry fuel. Fairbanks Morse had never attempted to adapt an engine for coal-diesel fuel operation, and the technology transfer from the earlier Cooper-Bessemer prototype coal-diesel engine was successfully completed.

- Demonstration tests were conducted at the engine manufacturer's test facility in Wisconsin, utilizing bituminous coal based CWF processed in Pennsylvania.

- Developed a path to market including specific milestones to commercialize the coal-diesel technology.

1. Process Description, Key Components, and Technical Issues

1.1 Coal-Diesel Power Plant System Description

The clean coal diesel represents a novel technology that has technical, economic and environmental advantages over conventional industrial scale power generating methods. This innovative technology enables utilization of coal-based fuel in large-bore, medium-speed, diesel engines. Modular power generating applications in the 10 to 100 megawatt size range are the target applications.

The technology involves the operation of a stationary diesel engine modified to operate on coal-water fuel instead of heavy fuel oil or natural gas which are the conventional fuels. The targeted performance characteristics of the mature commercial embodiment of the Clean-Coal diesel, once achieved, will make this technology quite competitive:

- 48% efficiency
- $1300/kW installed cost
- Emission levels controlled to 50-70% below New Source Performance Standards

A typical Coal Diesel System with a single 6 MW engine is shown in Figure 1. The Clean Coal Diesel System includes a multi-cylinder, medium speed diesel engine (either Fairbanks Morse or Cooper-Bessemer), generator, integrated emission control system and standard auxiliary systems. Engine exhaust will be fed to a waste heat boiler to generate steam for additional power generation and for power plant heating.

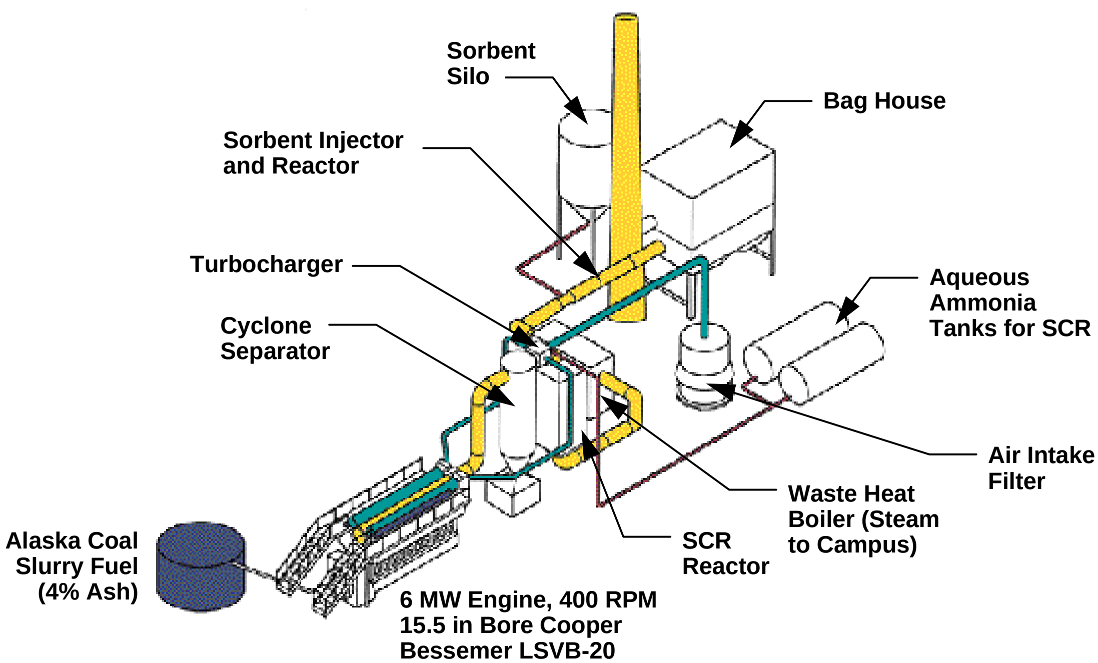


Figure 1. Coal Fueled Diesel Power Plant (6 MW)

The coal-diesel power plant is supplied with coal slurry fuel which is produced from raw coal feedstock at a regional coal-water fuel processing plant (analogous to an oil refinery). The process plant would utilize local coal brought by truck or train from coal mines within economic transportation distance. Figure 2 shows how two types of coal-water fuel will be produced (one for use in coal diesel-engines and a coarser one for use in modified oil boilers).

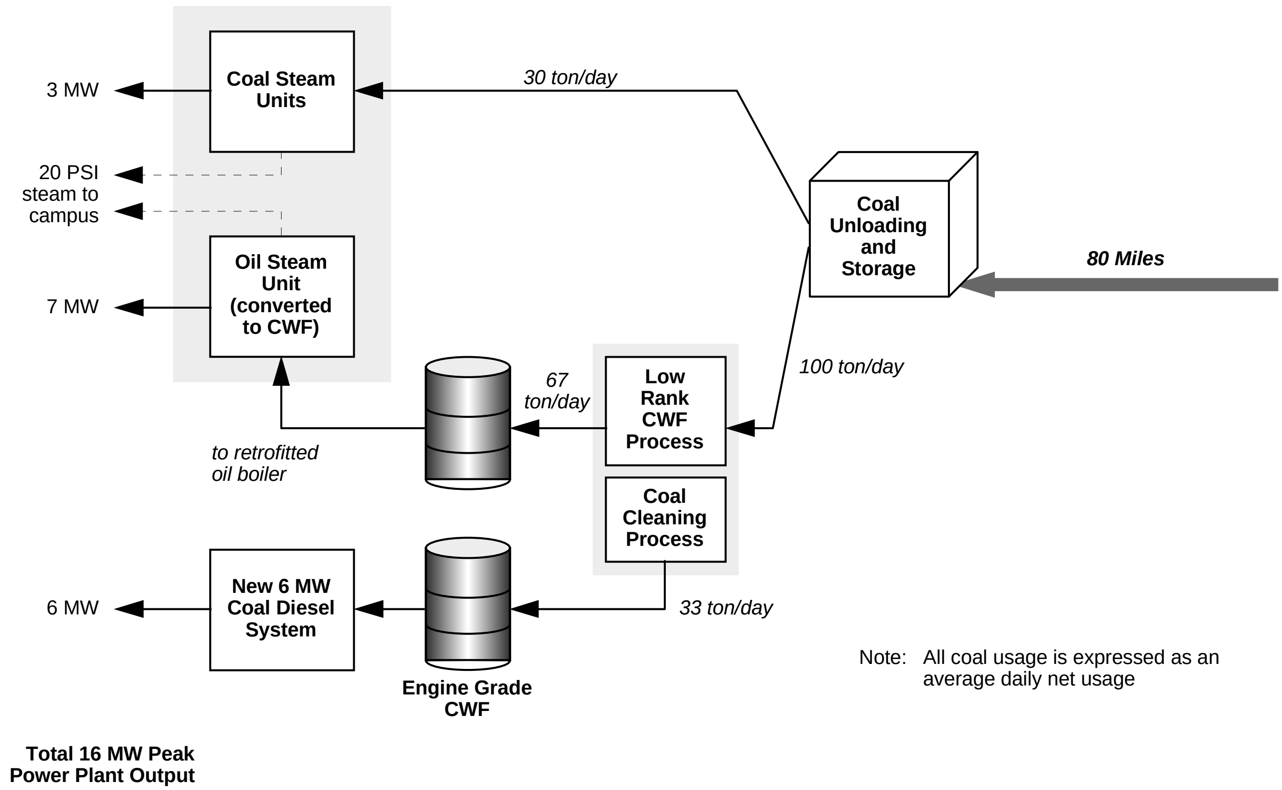


Figure 2. Schematic of Typical Combined use of Engine-grade and Boiler-grade Coal Slurry

The Clean Coal Diesel is intended to offer substantial benefits for industrial and municipal power plants:

- Rather than only being able to use relatively costly oil or natural gas for new generating capacity for plants of approximately 10-100 MW electrical power output, this technology would permit less costly coal to be used. The coal engine would be designed to switch over on the fly to gas or oil as well.
- The new fuel-flexible diesel engine would provide the additional capacity and provide for black start capability, eliminating power purchases during start-up.
- Reduction in the use of fuel oil by using more coal may reduce plant fuel costs. Today, the typical small industrial or municipal power plant experiences relatively high cost for fuel oil and natural gas.

1.2 Fuel Injection System for Coal-Water Fuel

1.2.1 General Injector Design Incorporating Shuttle

The normal diesel fuel injection pump and injector is a high-precision device operating with a close-tolerance plunger to produce high pressure pulses of known volumes of fuel at very precise timing. Since the coal-water fuel consists of hard particles that would cause erosion and seizure in any close-tolerance moving parts such as these, a novel approach was necessary to create similar injection events with coal-water fuel. Our technology included a shuttle piston operating on "harmless" diesel fuel which produced the dynamic volumetric displacement events required to inject the CWF. In other words, the use of a shuttle enabled us to operate the diesel pump ***indirectly.***

Figure 3 shows the general layout of the injection system, with ordinary diesel fuel filling all portions of the system except for the nozzle tip downstream of the shuttle. Each engine cycle, the predetermined correct volume of coal slurry was placed into the nozzle tip by a check-valve mechanism.

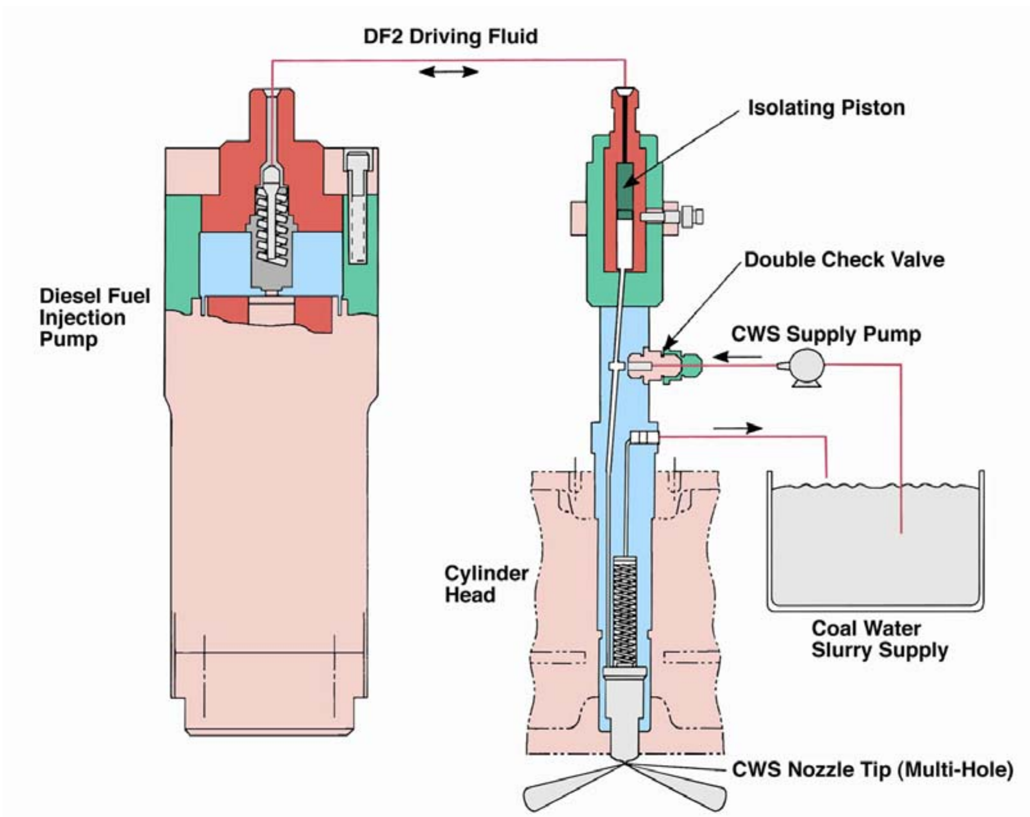


Figure 3. General Layout of CWF Injection System with Shuttle Piston

1.2.2 Injector Assembly for Fairbanks Morse Engine (Pielstick Model PC-2.6)

The CWF injector for the Pielstick 2.6 engine was designed around the envelope of the standard injector and the valve rocker arm mechanism. The final design required the injector shuttle body to be offset approximately two inches to clear the valve rocker arm mechanism for the ease of installation and removal of the injector. This injector was also designed to accommodate a Bentley Nevada proximity pickup to detect the movement of the nozzle assembly needle lift and a pressure transducer, manufactured by Piezotronics, to monitor the injection pressure at the top of the CWF shuttle. This injector as seen in Figure 4 and Figure 5 was different from the Cooper design in two distinct ways; the FME design was located under the engine valve covers where the Cooper design was extended to place the shuttle mechanism outside the valve cover. The other design difference was that the FME design incorporated an amplification piston for the internal oiler system which required an approximately 1/10th of the supply pressure of the of the Cooper design which did not have this feature. Noting the Cooper design required an air/oil amplifier to obtain the required oil internal pressure of approximately 2000 psi above the peak injection pressure during the injection cycle.

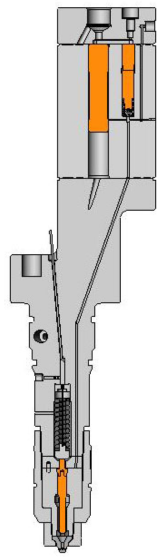


Figure 4. Injector Design for Fairbanks Morse CWF Engine

Figure 5. Main Coal-Water Injector and Pilot Injector

1.2.3 Shuttle Assembly

Generation I: The initial shuttle assembly was design for a capacity of 20K mm^3 to achieve the estimated require CWF quantity to obtain the required power requirement of the engine. The design capacity included additional capacity to accommodate any variance in the CWF additional fuel capacity required due to any variance in the CWF BTU content. This design included the internal oiler amplification piston system to obtain a lower oil supply pressure. This particular design feature was not included in the original Cooper. We had also the shuttle piston and the oiler pistons coated with tungsten carbide to minimize the wear of these components.

The design of sealing surfaces of the initial design where found to be marginal and appeared to be sensitive to the sequence in which we tightened the securing bolts that held the top plate and shuttle body to the injector main body. To eliminate any future leakage at these surfaces material was removed to achieve a higher unit loading at the sealing surfaces.

These changes in the pressure design of the shuttle body were found to be acceptable when retested during the bench testing at FME with DF2, and were also acceptable during later testing either on DF2 or CWF.

Generation II: Seizures of the shuttle piston which caused scoring of the shuttle piston and bore (see Figure 6 and Figure 7) were experienced during the initial phase of the bench testing of the injectors on the original CWF fuel at Fairbanks Morse Engine. These seizures had appeared to be attributed to a combination of factors: proper level of surfactant, proper oil supply pressure to the internal oiler or possibly a minimal diametric clearance between the shuttle piston and bore. The testing pointed to the CWF as it did not appear to discharge from the injector tip in a fluidic state during the switch over from DF2 to CWF, however, the other related items may have also contributed to the seizures.

To resolve these seizures the following steps were taken to achieve a higher success in the injection of the CWF without having any additional plunger seizures.

Figure 6. Shuttle Piston Showing Scoring

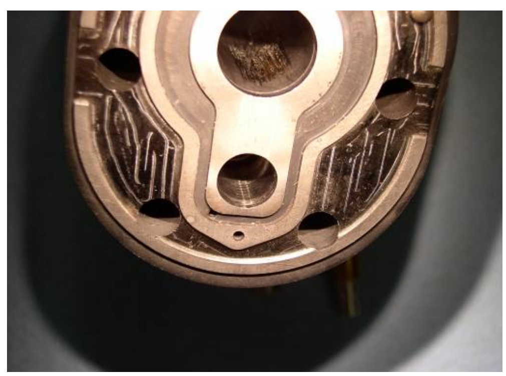


Figure 7. Shuttle Bore Showing Scoring

The shuttle pistons were remade and the shuttle bores were reground to remove the slight scoring of the bore. We had reviewed the clearances that had been used to fit the original shuttle to the bore, it was noted that the actual clearances were on lower side of the specified tolerance. Therefore, the new shuttle pistons were fitted to the upper end of the clearance, in order to achieve the following: a slightly higher leakage through the lower portion of the piston clearance to maintain a sufficient oil barrier and possible changes in the piston to bore clearance due to any distortion due to the clamping or due to heat.

These changes which were incorporated were proven successful, as we did not experience any further piston seizures once the proper CWF mixture, particle sizing and additive package was determined.

1.2.4 Needle Assembly

The comparison of the FME vs. Cooper design is as follows. The needle valve configuration was different, whereas the Fairbanks needle valve was of a larger diameter, which could possibly have more mass; the exact differences were not quantified. The FME needle valve upper portion is similar to that of Cooper needle valve, but its assembly is different. The Cooper design had a tool steel valve down to approximately 0.250 inches from the valve seat. The tungsten carbide valve seat was silver brazed into the tool steel upper body, which on numerous occasions separated during engine test. The Fairbanks design was reversed, where the lower portion of the valve up to approximately 0.375 inch is tungsten carbide and the upper portion, which is tool steel, is pressed on. This change was implemented to move the assembly joint away from the high-pressure area and to eliminate any separation during the injection event that was seen on the Cooper design.

The other difference was the method of assembly of the tungsten carbide seat to the nozzle body. The Fairbanks design has the seat press fit into the nozzle body, rather than having the seat silver brazed in place. This press fit design is to eliminate any movement of the seat, which would reduce the valve opening pressure that had occurred on the Cooper design on numerous occasions. This type of assembly method is common in the fuel-injection industry; however, assembly procedure is critical.

However, during the initial stages of the FME program, the needle valve stem had failed during bench top operation this fracture occurred at the base of the stem (see Figure 8). This failure had occurred on two of the needle valves. However, the noted damage that appeared to have resulted after the initial failure, made it difficult to determine the initiating point of this failure. The type of failure noted typically is caused by a machining, heat treatment or assembly error.

The stem failures that Fairbanks experienced may have been attributed to two factors, one from a bleed hole which was added for assembly and the possibility of cracks introduced at the base of the valve stem during the assembly procedure.

Upon review of the other FME needle valves, it was noted that a small vent hole was machined at the base of the valve stem through the radius. This vent hole, if not properly processed during and after heat treat, would result in a high stress concentration that could generate stress cracks during the finish grinding operation. These cracks were probably enhanced during the assembly operation of pressing the upper valve, which is made up of the valve stem and valve stop, onto the main valve. These observations would have caused the failure seen in Figure 8.

The design of the upper component of the valve assembly was changed by removing the orifice and the methodology of assembling the upper and lower components of the valve assembly. These changes were evaluated during several sequences of the bench testing of the CWF mixtures over a three year period prior to engine testing. It was noted that there were no subsequent failures in this area of the valve stem.

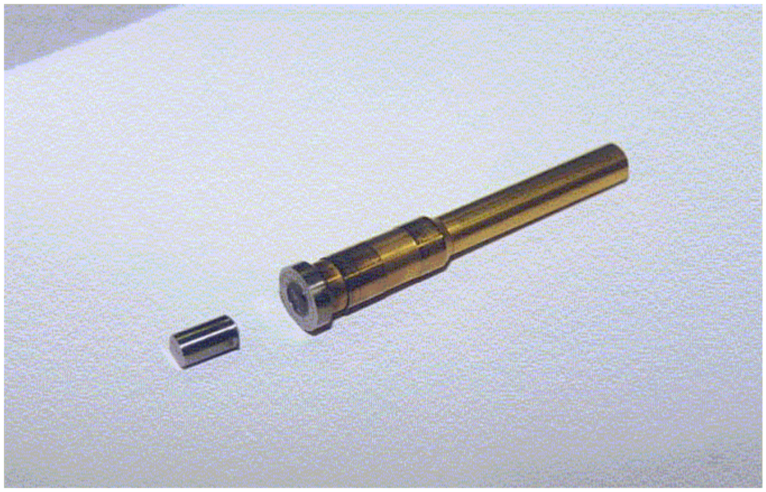


Figure 8. Photo of Needle Assembly

These above mentioned changes were successfully run on the engines during preliminary engine testing of the injectors on diesel fuel and the initial testing of the Pielstick engine on CWF.

1.2.5 Nozzle Body

The FME nozzle body was designed in concept similar to the Cooper nozzle design, but to fit within the parameters of the Pielstick standard nozzle tip. The other difference is the method of assembly of the tungsten carbide seat to the nozzle body. The assembly procedure incorporated into the FME design had the tungsten carbide seat shrunk fit into the nozzle body rather than having the seat silver brazed in place. This method of assembly is to eliminate any movement of the seat, caused by the softening of the silver braze in some instances, which would reduce the valve opening pressure that had occurred on the Cooper design on numerous occasions. This type of assembly method is common in the fuel injection industry; however, assembly procedure is critical.

The nozzle body cracking (see Figure 9) could have been caused by a variety of factors, an assembly problem, which possibly could have introduced a crack when the seat was pressed into the body or the combination of the shrink fit of seat into the nozzle body and higher injection pressure could have caused the eventual failure. It could also possibly be a heat treat problem, if there was not a sufficient case core definition in the area of the seat that could attribute to an eventual cyclic failure. The remaining possible cause of the cracking in the nozzle body could be a material related problem, which during heat treat could create a section that could be susceptible to cracking.

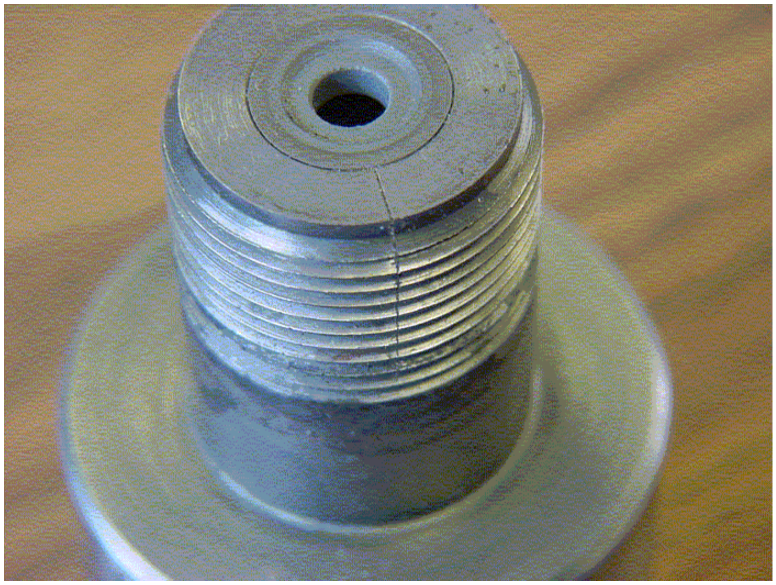


Figure 9. Photo of Nozzle Body

All the above possible factors that may have caused these failures were reviewed with the vendor who made the first prototypes. It was decided to redesign the area of the tungsten carbide seat to have the seat ground to the same level of the high pressure surface, allowing the use of a brazing method for the seat assembly to the nozzle body.

These design changes were evaluated during bench testing and were successfully run in the engine during the initial running on DF2 fuel and CWF.

1.2.6 CWF Check Valve

The CWF check valve design (Figure 10) was similar to that of the Cooper design with some basic design changes. The main change was the use of pre-heated steel instead of a heat treatable stainless steel for the other structure of the check valves. This change was made to strengthen the design in areas which failed when the fittings were torqued into place. In addition to the change in material, various wall thicknesses were increased in the previous areas of failure.

The check valves were made of tungsten carbide with increases in the diameters. Changes in the seat design were implemented to eliminate a possible feather edge during manufacturing.

This check valve design operated successfully during all stages and all phases of testing of the CWF injection system.

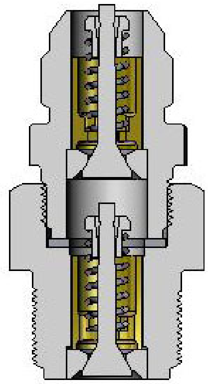


Figure 10. Check Valve for CWF

1.2.7 Hardened Nozzle Tips for CWF

The nozzle tips designed for the use testing of the CWF on the FME application were of two types. The initial button design (see Figure 11) was made of a sintered tungsten carbide material with a nickel binder. The initial orifice configuration was 10 orifices @ 0.65 mm which was similar to that used for heavy fuel for this FME engine application. However, after some preliminary bench testing with CWF it was found that the injection pressure achieved with this orifice size was too high. Therefore the orifice size was increased to 0.070 mm, which dropped the injection pressure into a safe operating range for the injection pump. This orifice configuration was successfully used throughout all bench testing and engine testing runs at FME.

The second design (see Figure 12) was the encapsulated sapphire design, which is similar to that used during the Cooper engine testing for a backup solution having the orifice size similar to that of the 10 orifices x .65 mm. This design was not tested on the bench or engine at FME during this test period. (It had been successfully tested at Cooper in 1996, but the design in Figure 11 was seen as preferable because there were no inserts).

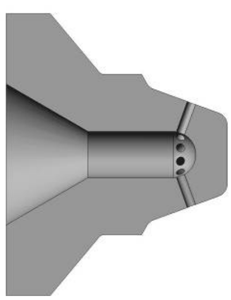


Figure 11. Monolithic Ceramic Button Injector Tip

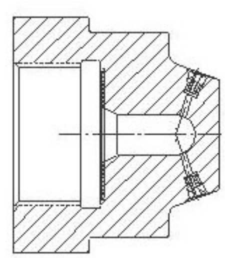


Figure 12. Encapsulated Sapphire Type Injector Tip

1.3 Durable Engine Parts for Coal-Water Fuel

The fundamental challenge of the coal-slurry fueled diesel is to protect the moving parts of the engine that are exposed to either the coal-water fuel (which is abrasive) or the solid particulate products of combustion which contain both ash and traces of unburned coal. By electing to use solid micronized coal rather than gasification, the intent was to gain significant cost savings. (Dedicated coal gasification facilities in the 10-50 MW scale are considered prohibitively costly). The challenge is to develop hard coatings and other material solutions which prove to be less costly over the life of the engine than the incremental cost of coal gasification and hot-gas clean up. (Note: until more extended demonstration time is actually performed, this goal is still speculative.) As shown below in Figure 13 and Figure 14 there are several specific engine components that need protection:

- Fuel injection pump system and nozzle tip
- Piston Rings and Liners
- Exhaust gas valves and seats
- Turbocharger rotors and blades
- Crankshaft bearings need protection from any used oil contaminants (e.g. ash)

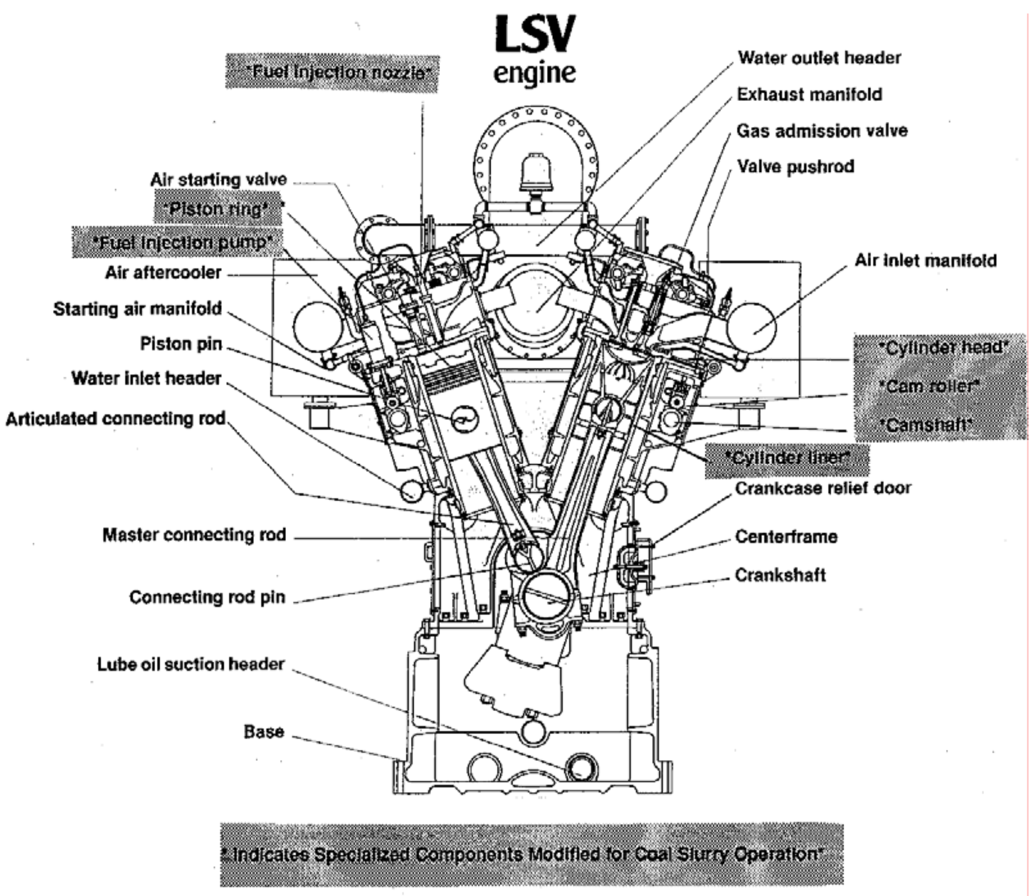


Figure 13. Highlights of Novel Coal-Fueled Diesel Engine Technology Elements

Exhaust Valves: Through the test series completed in 1996, Cooper accumulated durability experience on the two chromium carbide exhaust valves installed in the #1 (CWS) cylinder on the Cooper LSC engine. The valves accumulated 180 hours of operation (91 hours on CWS). The CWS operation consumed slurry which had an ash content between 2.0 and 2.7% by weight. Engine operation was at rated speed (400 rev/min) and covered a range of loads from 150 to 208 psi brake mean effective pressure. Much of the operation was at high load.

Component	Wear Mechanism(s)	Best Solution Tested
Injection Nozzle Tip Orifices	Solid particle erosion and cavitation	Ceramic or metal oxide inserts
Injection Nozzle Valve	Solid particle erosion	Hardened steel alloy; monolithic ceramic
Injection Nozzle Valve Seat	Solid particle erosion	Monolithic ceramic
Injection Nozzle Shuttle	Solid particle erosion	Hardened steel alloy

Component	Wear Mechanism	Best Solution Tested
Cylinder liner	Three body abrasion	Monolithic ceramic plasma coating
Top compression rings	Three body abrasion	Monolithic ceramic coating
Other compression rings	Three body abrasion	Monolithic ceramic coating
Oil control rings	Three body abrasion	Alloy
Exhaust valves	Solid particle erosion	Monolithic ceramic coating
Exhaust valve seats	Solid particle erosion	Monolithic ceramic coating

Figure 14. Wear Mechanisms and Hardened Part Technology

Valve Inspection

Coating Description. The chrome plated stems of two exhaust valves were surface ground to remove 0.015" of the plating prior to chromium carbide coating with Praxair Surface Technologies' "LC-1H" coating. Valve and coating material characteristics are described in

Table 1 and Table 2. The coating was applied by a detonation gun in which the gun is stationary and the part is both rotated and translated in front of the gun. This process results in a very uniform coating thickness over the entire part. The initial coating thickness was 0.021/0.023”. The valve stem was ground and polished to a coating thickness of approximately 0.015”, yielding a shaft diameter of 0.993/0.994”. The valve seat was also ground and polished to a surface finish of 9-11 microns, removing only 0.003” of coating.

Wear Performance. The exhaust valves were inspected before and after the 90-hour CWF test was conducted. The initial and final dimensions of the valves are summarized in Table 3. The results indicate that the coated part experienced little or no wear. For example, the diameter of the stem remained within the original tolerance of 0.993/0.994” (refer to Table 3. column TC1 and TC2: row D-12, D-14, and D-16). Furthermore, the valve dimensions at 17, 17.5 and 18” from the valve end opposite the head remained the same (refer to Table 3. TC1 and TC2: row D-17, D-17.5 and D-18). Figure 15 shows a slight build-up of reddish-brown material on the edge of the valve head. Figure 16 shows the same material on the valve head as well as well-defined and evenly spaced wear tracks cutting through the debris. This small amount of material deposited on the valve is easily removed by light scraping and is probably coal ash. The circumferential wear tracks results from valve rotation (valve rotators were used). The deposits did not appear to affect engine operation. Figure 17 shows the surface texture along the length of the valve. The absence of wear on these coated valves after 90 hours of CWF operation is a dramatic improvement compared to past results.

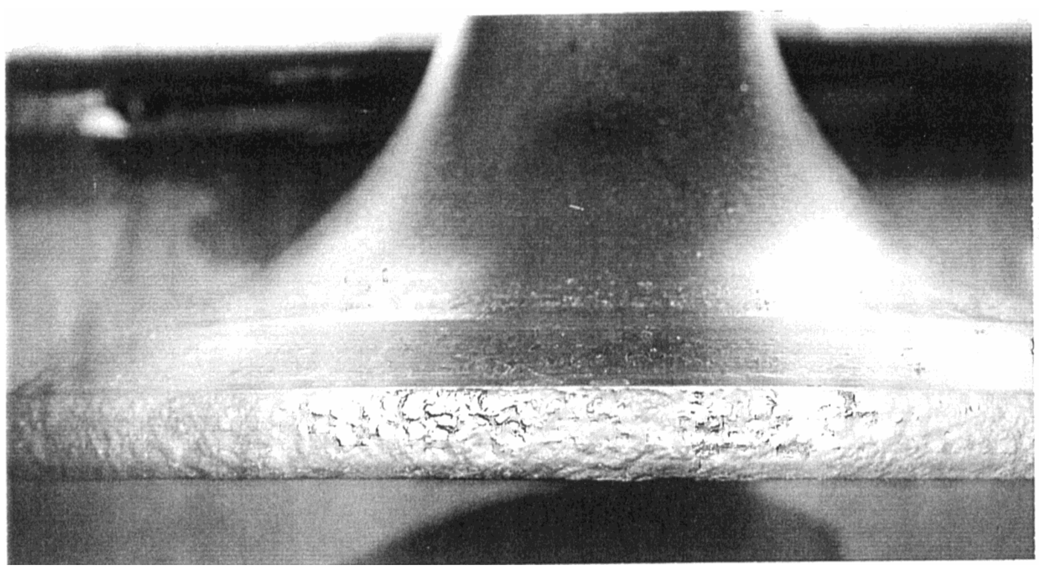


Figure 15. Edge of Exhaust Valve Head

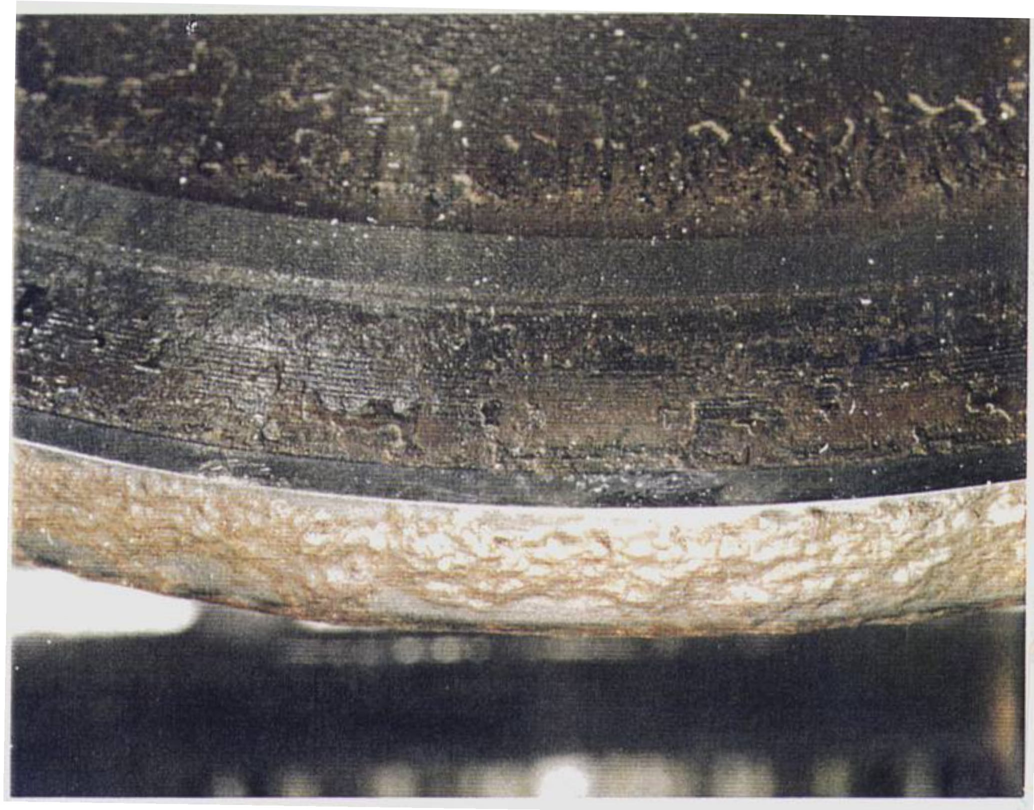


Figure 16. Exhaust Valve Head

Figure 17. Surface Texture Along the Length of Exhaust Valve

Table 1. Valve Materials Characteristics

Section*	Material	Heat Treatment	Hardness R_c	Notes
Head	Inconel 750	1500°F for 24 hours and air cool	30 (minimum)	
Stem	Alloy Steel AISI-3100	Liquid Quenched and tempered	25 minimum 36 maximum	Chrome-plated**

*Note: the head and stem are welded together and stress relieved at 1100°F minimum for 2 hours. The weld is checked by fluorescent penetrant inspection.

**Chrome plating is removed before chromium carbide coating is applied.

Table 2. Coating Material Characteristics

Coating Designation	Principal Constituents	Nominal Composition (wt. %)	Hardness (Rc)	Hardness (HV.3) (kg/mm2)
LC-1H	Chromium Carbide	80% (92Cr-8C) + 20% (80 Ni-20Cr)	64	775

Table 3. Exhaust Valve Dimensions Before and After 90 Hours of CWF Operation

		TC1 = Left Valve		TC2 = Right Valve	
Location, inches	**Print**	**Initial (0 hrs.)**	**90 hrs.**	**Initial (0 hrs.)**	**90 hrs.**
D-12	.993/.994	0.994	0.994		0.994
D-14	.993/.994	0.994	0.994		0.994
D-16	.993/.994	0.994	0.994		0.994
D-17		1.116/1.121	1.121	1.091/1.096	1.090
D-17.5		1/503/1.508	1.504	1.435/1.440	1.437
D-18		2.367/2.374	2.374	2.200/2.225	2.236
D-3	4.534/4.554	4.519/4.525	4.573/4.586	4.521/4.525	4.559/4.563

In-Cylinder Components Inspection

A number of in-cylinder components were inspected and measured for wear after Cooper completed a key segment of the Phase I CWS tests (which ran from August 1995 to May 1996).

Piston Rings

The top three compression rings for this test series were tungsten carbide coated rings which had already accumulated 100 to 230 hours of CWF operating time from previous tests. After 121 hours of additional CWF operation, much of which was at full load [200 to 208 psi brake mean effective pressure (bmep)], these rings experienced only 0.007 to 0.014 inch end gap increase, as shown in Table 4.4. For comparison, a new standard ring (not coated with tungsten carbide) was installed as the 4th compression ring and it experienced a 0.148 inch end gap increase. The end gap increase would likely have been 2 to 3 times greater for the standard ring if it were installed as the top compression ring. The wear rates for the tungsten carbide coated rings were one to two orders of magnitude less than standard material piston rings.

A special chrome plated oil control ring was installed in the piston as the bottom ring for evaluation. Its initial wear rate was approximately 20 times less than the standard oil ring.

Table 4. Preliminary Ring Wear Data

Ring Position	Ring Type	CWS time on ring at start of test series (hrs.)	End gap increase after this test series (121 hours on CWS) (in.)
Top compression	WC coated	183	0.014
#2 compression	WC coated	232	0.010
#3 compression	WC coated	103	0.007
#4 compression	Standard	new	0.148
Top oil control	Standard	new	0.484
Bottom oil control	Chrome plated lands	new	0.028

Piston

There was no significant wear measured on the piston or piston skirt. A few vertical scratches were observed but no hot spots on the piston were seen. The ring lands showed a slight increase in size -- approximately 0.010 to 0.012 inch for the compression ring lands and 0.003 to 0.009 inch for the oil control ring lands.

Nozzle Tip

There was no significant wear measured on the sapphire inserts in our CWF nozzle tip. Nozzle hole wear was measured with wire gages. The starting hole diameter was 0.6330mm -- a 0.025 inch wire gage (0.635mm) would not fit any of the 18 inserts. This nozzle tip now has approximately 263 hours of CWF operation.

Durable Component Findings

Arthur D. Little materials specialists visited Cooper's engine laboratory to inspect in-cylinder components after Cooper completed a key segment of the Phase I CWF tests. Their findings are summarized below:

Exhaust Valves

The most recent set of chromium carbide coated exhaust valves appeared to be wearing at a low rate during the first 90 hours of CWF operation. This was a substantial improvement compared to the earlier first set of coated valves or the standard valve material. However, additional operating experience is needed to develop component life estimates.

Piston Rings

The top three piston compression rings for this test series were tungsten carbide coated and have accumulated 220 to 350 hours of CWF operating time. The overall wear rates for these test rings, based on end gap increase, was very low compared to conventional ring materials (one to two orders of magnitude lower overall wear rate). The compression rings were showing some signs of material distress, as revealed by the loss of coating in some areas and cracking in others. The cracking was probably due in part to rings with coatings that are too thick and therefore susceptible to cracking. Additional improvements in material integrity and ring life may be obtained by optimizing the coating process.

A special chrome plated oil control ring was installed in the piston as the bottom ring for evaluation. The chrome plated oil control rings showed wear resistance substantially greater than the resistance provided by standard rings.

Cylinder Liner

The tungsten carbide coated cylinder liner was in excellent condition after these 263-hr tests.

Nozzle Tips

The nozzle tip hole diameter (nominal size = 0.633 mm) increased less than 0.002 mm (< 0.3%) based on wire gage measurements taken after 263 hours of CWF operation. Detailed microscopic evaluation of each sapphire orifice showed some signs of chipping at the discharge side and erosive wear along the inner surface. Although the degree of wear appeared to have increased since the earlier detailed inspection, it did not appear to affect important overall engine performance measures such as fuel consumption or emission rates.

1.4 Suitable Coal Feedstock, Coal Slurry Process Plant Designs, and Additives

Coal-Water-Slurries (CWS) are two phase mixtures that generally exhibit complex flow and combustion characteristics. Important characteristics of these slurries include acceptably slow settling properties of the solids, tolerance to flow at high shears normally encountered in diesel injection systems, adequate atomization and evaporation properties of slurry sprays, and small enough particle size for ignition and combustion at compressed hot air conditions normally achievable in diesel engines. These properties of the slurry are dependent on the coal rank and source, the particle size distribution of the solids in the slurry, the mass loading of the coal, and the types and concentrations of additives that are used to improve stability, compatibility, and flow at high shear rate.

The specifications for suitable coal-slurry fuels derived from Cooper/Arthur D. Little engine test evaluations are summarized in Table 5. (75) In general the primary consideration is that the source coal must be low-ash and amenable to physical pre-cleaning by methods described below.

Table 5. CWS Properties Tested

Effect	Coal Property	Range Tested 3/28/90	Results
Combustion Performance	Volatile content Rank Heating Value Particle Size	27-41% Bituminous/subbit. 12-15 kBtu/lb (dry) 10-85 μm top size 3-20 μm mean size	All Okay
Emissions Control Cost	Sulfur Nitrogen	0.7-1.0 1.2-1.8%	<2% Okay TBD
Handling	Solids Content Viscosity	48-55% 200-400 cp	All Okay
Wear, CWS Cost	Ash content Hard Mineral Content	0.5-3.8% --	<1.8% Okay TBD

As shown in Table 5, particle-size distributions that have mean sizes in the range from 3 to 20 microns, and top sizes up to 85 microns are acceptable.

Both bituminous (Virginia) and sub-bituminous (Alaska) coals were successfully burned in two coal-diesel engines as part of this demonstration project. The choice between coal ranks is open, except that the heating value does have an impact on the injection requirements, and must be considered when designing the injection system. In an earlier GE project the sub-bituminous coals generally burned slower than bituminous coals. It should be noted, however, that the heating values of these fuels are lower and the injection rates were not adjusted in the GE tests to account for the lower rates of heat addition to the engine.

1.4.1 Pre-Processing of Bituminous Coals

Cleaning technologies that were examined in this program included heavy media separation, coarse flotation, fine flotation, oil agglomeration, and chemical cleaning. Several different approaches to each of these techniques are possible, but most of the coal slurry fuel used in this demonstration was cleaned by heavy-media separation.

Physical coal cleaning (see Figure 18), includes size reduction to ¼ inch by 200 mesh. The physical coal cleaning is accomplished by heavy media separation. This step produces engine-grade product (2-5% ash) and a byproduct middling stream of 8% to 10% ash. Once the coal has been cleaned, it is then metered, along with water, to a ball mill where it is reduced to approximately 250 microns. The final step is micronizing (fine grinding) to below 20 microns mean size and formation of the slurry.

In an earlier project GE (19) examined three different sources of bituminous and two different sources of sub-bituminous coal. Blue Gem Seam of bituminous coal, chemically cleaned by Otisca to less than 1 percent ash and sulfur, was tested in two different sizes and with two different additive packages.

1.4.2 Sources of Suitable Bituminous Coals

In general the primary consideration is that the source coal must be low-ash and amenable to physical pre-cleaning. A survey of suitable U.S. bituminous coal feedstocks was performed by Amax and state-by-state results and recommended coal seams are presented in Appendix E. Considering all known bituminous coal sources, perhaps 15% to 25% of these seams are suitable for processing into engine grade coal-water slurry. The majority of bituminous coal is too high in ash content. There is not much difference in the burnout and burn rates of the chemically cleaned bituminous coals. A physically-cleaned bituminous coal from one particular source (Kentucky Splint Seam) appeared to have a lower burnout than chemical cleaned coals.

1.4.3 Pre-Processing of Sub-bituminous Coals

In order to make coal-water fuel from Alaska coals (which are more reactive low-rank coals), it was necessary to use a continuous hydrothermal coal treatment process. One such version is known as hot water drying (HWD). The technical feasibility of producing moderately loaded CWF, with a dry solids content of around 60%, from Alaskan low-rank coal without the use of costly additives was demonstrated in a 7.5 tpd pilot plant at the Energy and Environmental Research Center in 1990-91. Alaska CWFs were shown to be premium fuels superior to heavy fuel oils in a pilot-scale boiler giving essentially complete carbon burnout. Sulfur dioxide (SO_2) emissions were well below compliance levels because of the remarkably low sulfur content of most Alaskan coals.

The Alaskan coal used in this project was pre-processed by hot-water drying (Figure 19). Processing of the coal was performed by EERC using the same process which had been designed and run at pilot plant scale by EERC in 1985-1988. Similar thermal hydrotreatment processes have been demonstrated by over twenty different investigators since the 1920's. In this process, the coal water mixture is fed through a series of reactors that ensure proper residence time, the proper temperature (approximately 280 to 300°C) and pressure conditions to convert the raw coal slurry to hot water dried coal.

Following the hot-water drying, EERC proceeded with the additional steps of fine-grinding and formulation of a coal-water fuel. That portion of the CWF intended for diesel engine fuel is subjected to fine grinding to 10 microns mean size (65 microns maximum).

A remaining barrier to widespread commercial applications of this technology is operation at a commercial scale to obtain realistic process economics data and thus enable potential CWF users to validate the process. This was an initial objective of the coal-water fuel plant portion of the CCT demonstration project at UAF, but funding restrictions did not allow this plant to be built and demonstrated. As part of this project a full detailed design of a Hydrothermal Coal Process Plant was completed including parts list and P&IDs (see Figure 20). The key module in the process is the hydrothermal treatment reactor, a photograph of which is shown in Figure 21 (this was the actual reactor used in this project at EERC).

1.4.4 Sources of Suitable Sub-bituminous Coals

Any low ash, cleanable sub-bituminous coal is suitable, such as most Powder River Basin coals and Alaskan sub-bituminous coals. The source coal in this project was supplied by Usibelli and is the same low-rank coal as is currently supplied to the UAF for boiler use in their power plant. This coal, depending on the specific mining pit, ranges from 7 to 9% ash and 0.1 to 0.5% sulfur, with a heating value of approximately 8500 Btu/lb (with 20-25% moisture).

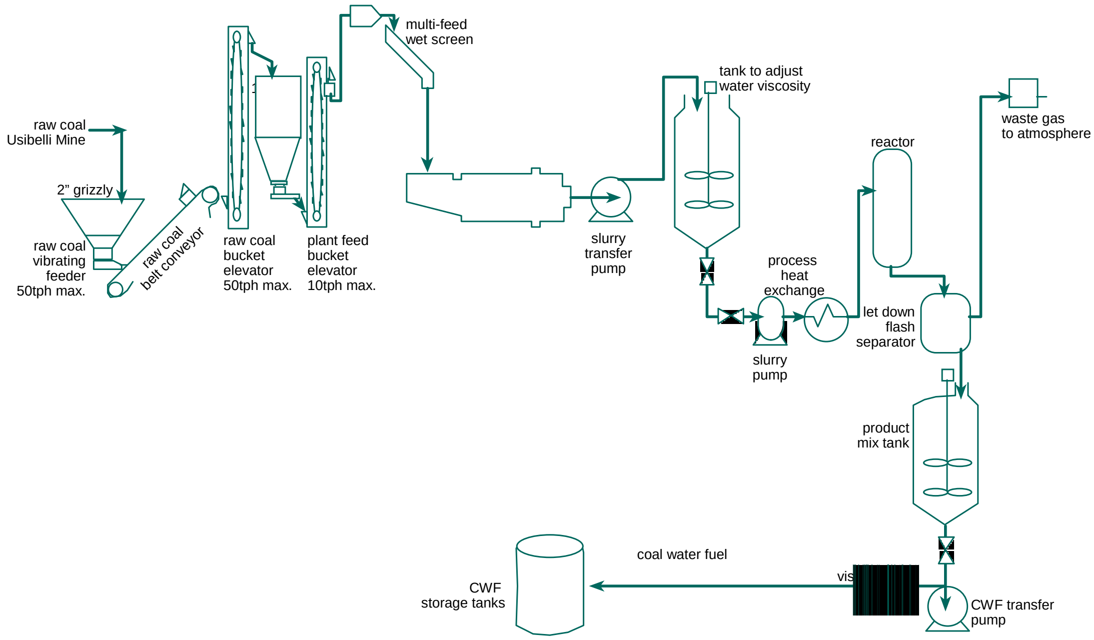
The Alaska coal fields comprise the largest accumulations of low-sulfur bituminous and sub-bituminous coal in the world. The coals here generally average less than 0.5 percent sulfur, but may contain values as low as 0.1 percent. Analyses of Alaskan coals indicate that they nearly all fall into the low-sulfur category. The coal of the Beluga field of the Cook Inlet-Susitna province, which have the lowest sulfur range of any U.S. coal (<0.1-0.3 percent), meet the Environmental Protection Agency's emission standards for direct combustion. The Nenana basin contains 8 billion tons of identified resources with total sulfur averaging 0.2 percent.

Table 6 provides the demonstration project's coal processing requirements (input and output) for 6000 total hours of coal diesel engine operation. When processing the source coal into engine grade slurry, rather than just make the engine grade product, it is more economical to produce two grades of coal slurry (a coarser grade for boilers and a fine grade for engines). Hence the "yield" of 33% engine-grade product is cited in Table 6. The balance of 66% of the coal goes to boiler grade fuel at much lower cost and price point.

Table 6. Coal Fuel Requirements for 6000-hour Demonstration (Assumes 2 Ton/hr Clean Dry Coal Input to Engine)

Year	At 33% Yield, Input Coal (tons)	Hours of Engine Operation	Clean Dry Coal Usage by Engine (tons)	Physical Cleaning Plant Operation to Produce Engine-CWF Hours (at 3.4 ton/hr output)	Hours to Process Reject Coal (into CWF for boiler use)
1	3400	500	1000	300	600
2	10300	1500	3000	900	1800
3	27500	4000	8100	2400 (1 to 2 ratio)	4800 (1 to 2 ratio)
Total	41200 (tons input coal)	6000	12100	3600 (for engine grade HWD coal slurry)	7200 (for boiler-grade HWD Coal Slurry)

*This is full load or maximum coal input. We assumed that part-load engine-operation would reduce the <u>average</u> coal input to 1.8 ton/hr



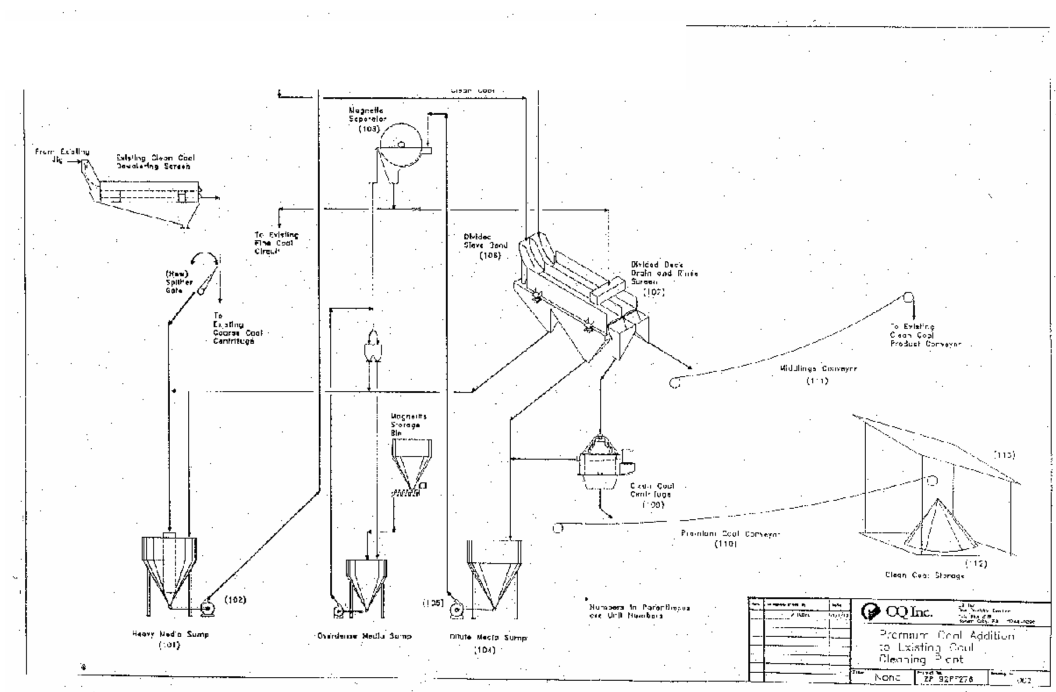


Figure 18. Physical Cleaning Process (CQ, Inc.)

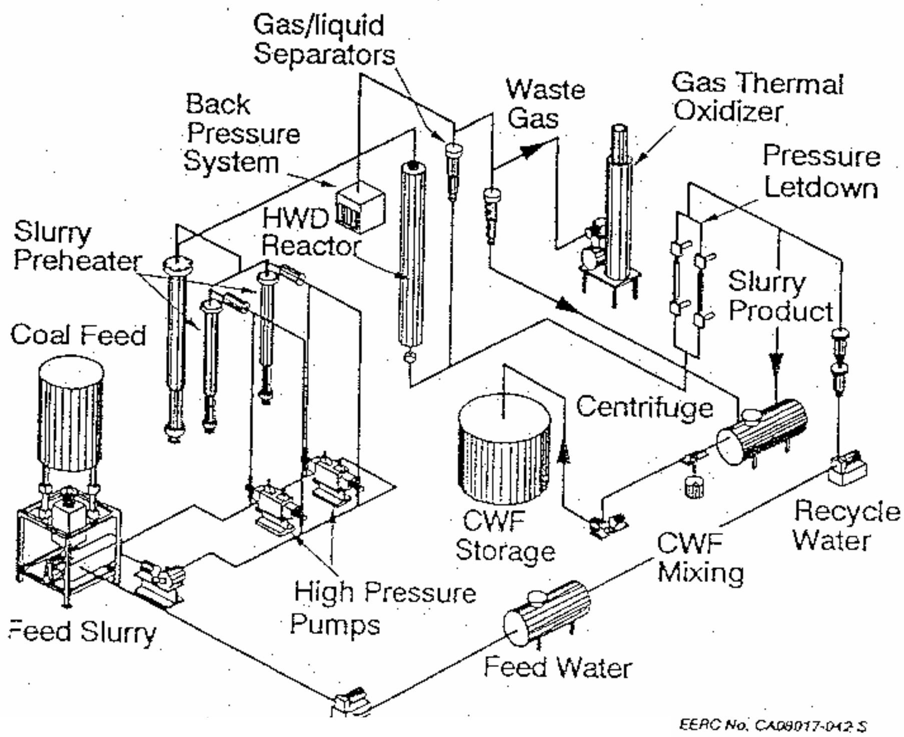


Figure 19. Hot Water Dry and Slurry Process

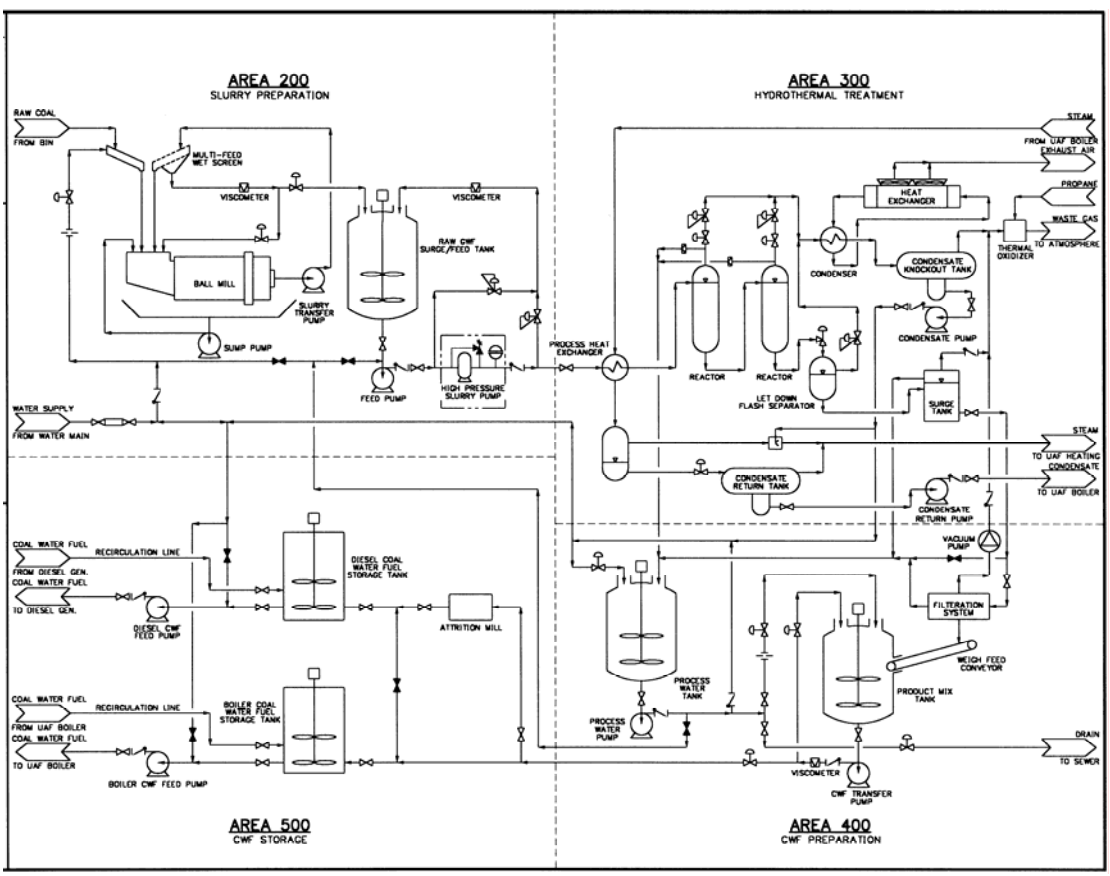


Figure 20. Hydrothermal Process for Low-Rank Coal

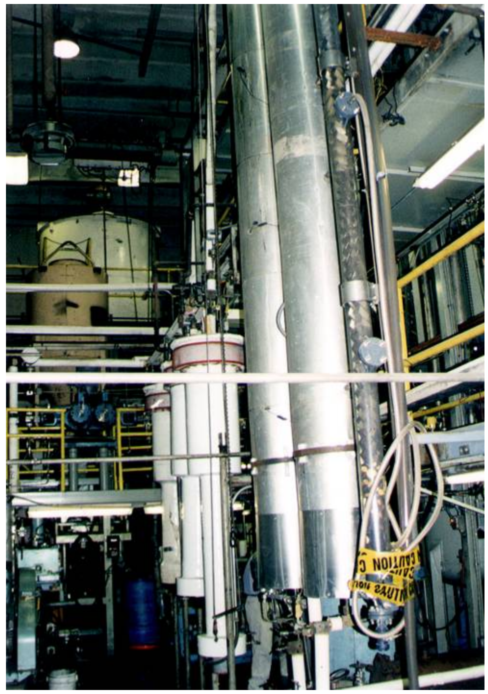


Figure 21. Hydrothermal Process Reactor at EERC (285C for ten minutes)

1.4.5 Additive Package for Coal Water Slurry Fuel

Additives were necessary to control the low and high shear viscosity of the slurry, such as xantham gum and surfactant, respectively, and dispersants to prevent agglomeration. Small amounts of each additive were adequate (on the order of 0.5% to 1% each). Appendix G summarizes the series of tests used to develop the additive package.

1.5 Emission Control System

1.5.1 Emission Control Subsystems for NOx, SOx, and Particulates

Effective controls for NO_x, SO_x, and particulate emissions will be essential for the successful commercialization of stationary Cooper-Bessemer coal-fueled diesel engines. We have established emission control system performance targets based on the projected needs of 10 to 100 MW cogeneration and independent power production sites in the year 2010 to 2030 timeframe. Table 7 summarizes the emissions targets (based on utilization of Alaska Coal) and the control methods that will be implemented to reach these levels. The Clean Coal Diesel will include an exhaust gas treatment system, including a cyclone, selective catalytic reduction (SCR), sorbent injection for SO_x control, a baghouse and new exhaust stack to assure appropriate control and dispersion of air emissions. During the prior DOE-METC funded development program, a full scale emission control system sized for an 1800 kW coal diesel engine was demonstrated to be capable of meeting all of these performance goals. This system was transported and re-commissioned at Fairbanks Morse, and was operated as part of the demonstration project in 2003-2004. The system is shown in Figure 22 and Figure 23 including the baghouse and SCR Unit.

Table 7. Emission Control Target Levels (Alaska Coal)

Pollutant	Control Methods	Emission Target
NOx	• Water Injection (CWF) • Combustion Optimization • Selective Catalytic Reduction • Dry Sorbent Injection	0.15 lb./MMBtu
SOx	• Coal Cleaning • Dry Sorbent Injection	0.12 lb./MMBtu
Particulates	• Cyclone • Baghouse	0.08 lb./MMBtu

When operated on coal fuel as designed, the UAF coal-fueled diesel operation was expected to result in significant reductions to overall annual emissions from the UAF power plant. In any extended period of maximum coal-diesel utilization, the four criteria pollutants (SO_2, NOx, particulates and CO) are estimated to be 35-50% lower than emission levels that would be experienced without the coal-diesel in operation.

As part of the demonstration project, an engineering company prepared detailed specifications and purchased components under competitive bidding for the coal diesel's emission control system that was installed at UAF. Included in the design document delivered to DOE were conceptual arrangements, heat and material balances, and performance requirements for the silencer, cyclone, SCR reactor, sorbent injection system, and baghouse. This information provided the basis for the A&E team's preliminary and detailed designs.

Figure 22. Coal Diesel Emission Control System (Baghouse)

Figure 23. Coal Diesel Emission Control System (SCR Unit)

1.5.2 Special Precautions Required for Bag-house Protection

On April 14, 2004, after several months of set up and check-out tests, a coal-water slurry mixture was successfully injected and combusted in a two-cylinder test engine at Fairbanks-Morse Engines Test Facility located in Beloit, WI. The fuel consisted of EERC-prepared CWF which had been processed from Usibelli coal (an Alaska sub-bituminous coal). This fuel was successfully fired in the engine for close to an hour at 17% load, then the engine was switched back over to DF2. Following the switchover, the engine was again switched over to CWF for another 9 minutes run time.

This test run on coal fuel was the initial attempt to burn coal slurry to see if fuel-injection and pilot ignition systems were functioning properly for good combustion, and to then to make adjustments (tune the engine pilot timing etc) accordingly. We observed good combustion in one of the two cylinders ("right" hand side of the engine) but the other cylinder had incomplete combustion and emitted some partially burned coal particles (still glowing). Towards the end of CWF run, there was a baghouse fire attributed to these glowing unburned coal embers in the exhaust, which occasionally occurs in new gasification plants during shakedown testing. The "sparklers", as they are called, were emitted from the exhaust valve of the engine, carried along in the gas stream, and ignited the baghouse bags. A time history trace is shown below in Figure 24. The suspected onset of the fire is shown as the peak in the baghouse inlet temperature. The differences in exhaust temperature is explained by the differences in combustion temperature between the two cylinders on CWF—the left cylinder combustion continued through the Exhaust Valve Opening event, so apparently some combustion continued through the exhaust system. This late combustion was due to the phasing of the pilot injection particularly in the "left" cylinder, which during this inaugural run was too late, causing the CWF combustion to initiate too slowly. Most likely this was a major contributor to the ignition of the baghouse fire

The root causes for the baghouse fire are from a combination of the following four problems (all of which could be corrected):

(a) Imbalance in combustion between the two cylinders. The ignition in the "left" cylinder needs to be tuned and improved to match the good slurry combustion in the "right" cylinder.

(b) Late DF2 pilot injection during the coal water slurry run. This caused a delay in start of the CWF combustion, which in turn led to residual coal particles carried by the exhaust gases, which were burning through the exhaust valve opening phase (EVO) and produced "sparklers."

(c) Insufficient Cooling of Exhaust Gas. The fan installed at Fairbanks Morse Engine was inadvertently undersized for the throughput, as well as for the gas pressure drop encountered during the testing. This can be readily corrected.

(d) Lack of "Sparkler" suppression. During the earlier Cooper runs in 1996, a cyclone was installed between the engine and the turbocharger to remove the large particles, but was known to have poor collection efficiency and thus was not installed at FME. Thus, in the current configuration, "sparklers" were allowed to pass through to the baghouse.

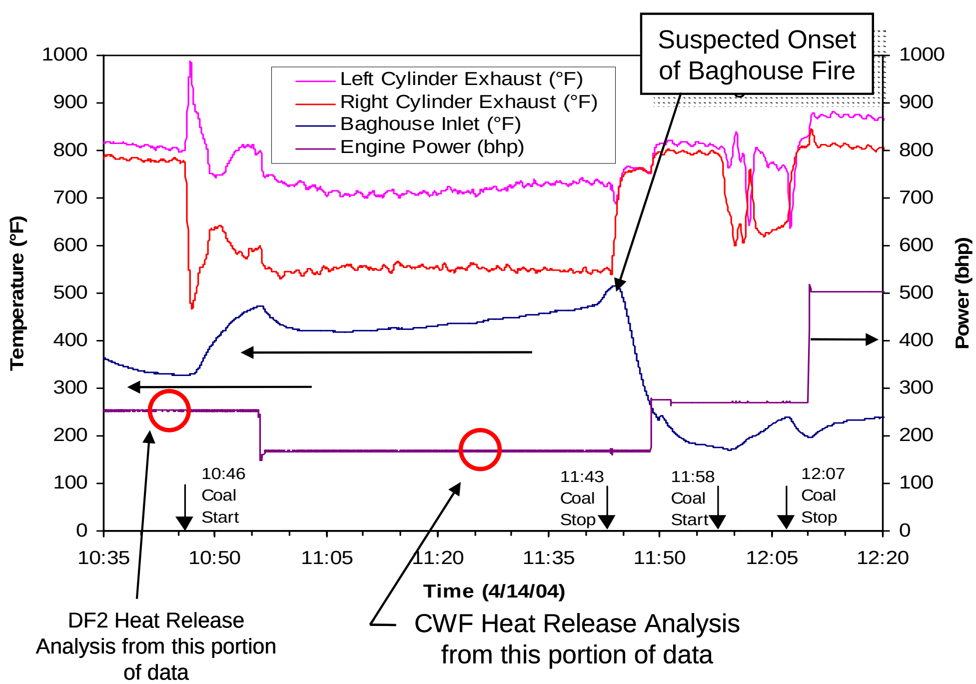


Figure 24. Time History of April 14, 2004 CWF Run

Solution: TIAX recommended the following solution to the baghouse fire as a three-path effort to address the three root causes.

(a) To address late combustion: For root cause #1, the combustion timing will be adjusted such that the pilot injection ends shortly after the beginning of CWF injection. This will cause an initial pre-mixed combustion spike, igniting the CWF quicker, thus burning more CWF before EVO.

(b) To address insufficient cooling: The second root cause of the baghouse fire is the insufficient cooling of the exhaust gas. The manner in which the exhaust gas is cooled currently, as well as, the proposed solution is shown in Figure 25.

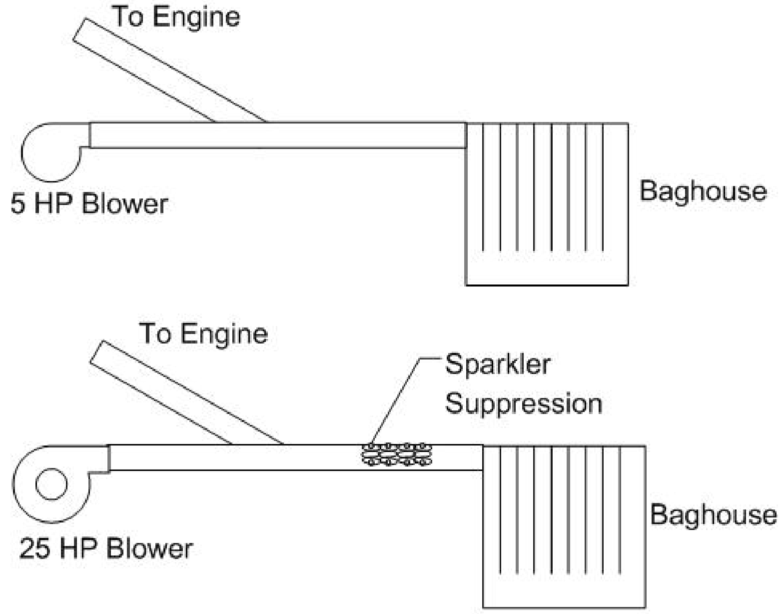


Figure 25. Schematic of Current and Future Exhaust Cooling Systems

The current 5 hp blower was rated at 6,675 SCFM @ 2 inches w.c. During the engine operation, the turbo outlet pressure was recorded to have a maximum of 11 inches w.c, which greatly reduced the throughput of the current blower and thus the cooling capacity of the blower, which dilutes the exhaust with ambient air. The upgraded blower is rated at 5,040 SCFM @ 13" w.c. and provided adequate dilution air to cool the exhaust stream.

(c) To address sparklers: The third root cause for the baghouse fire was the lack of "sparkler" suppression. A simple method that has been used in the past for gasification plants was the suspension of chains in the exhaust stream to create a torturous path for any glowing embers of ash. The chains are offset in such a manner so as to completely block the passing embers, but still produce a negligible pressure differential through the entire chain set. A schematic of the chain system is shown below in Figure 26.

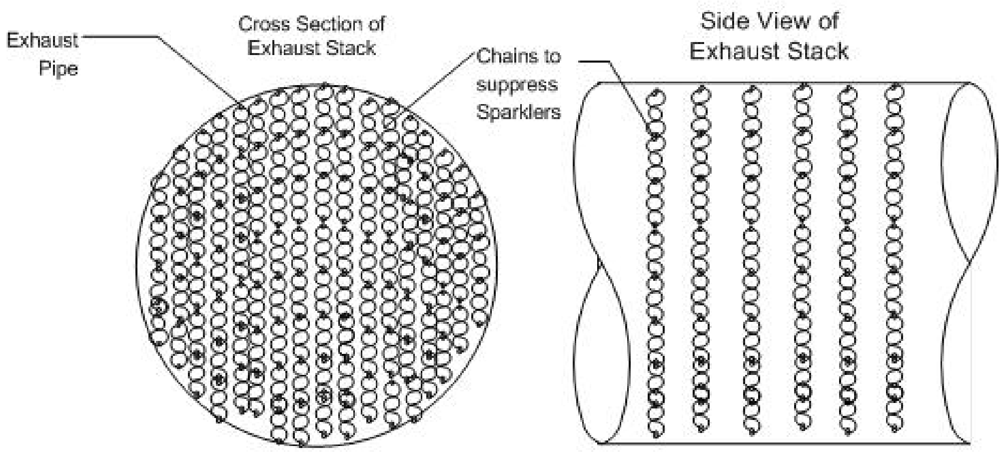


Figure 26. Schematic of Sparkler Suppression

2. Coal Diesel Power Plant Installation at University of Alaska

The University of Alaska campus in Fairbanks, Alaska, was the project's host site. At this location, during 1998-2000, the University constructed and commissioned the Coal Diesel System, which now (after the project) serves as a 6.2 MW diesel power plant addition (to the existing two oil-fired boilers and two stoker-type coal-fired boilers). Figure 27 provides a plan view of the complete project at the UAF campus site. The Clean Coal Diesel System included an 18-cylinder, 512 rpm Fairbanks Morse PC-type engine, generator, integrated emission control system suitable for coal operation and standard auxiliary systems. The engine included an extra set of hardened parts (e.g. piston rings) required for coal operation. Engine exhaust was fed to a waste heat boiler to generate steam for campus heat.

R. W. Beck was the A&E for the design-construct project. As is required by clean-coal demonstration capital projects, the design and construction phase included milestone reviews at both the 65% design point and the 95% design point. The final capital cost exceeded the preliminary design estimate (based on an 18-cylinder diesel engine) by about 10%. This growth was largely attributed to higher than expected bids for the HRSG, SCR and baghouse, and to inclusion of electrical equipment in the coal processing plant that had been omitted previously.

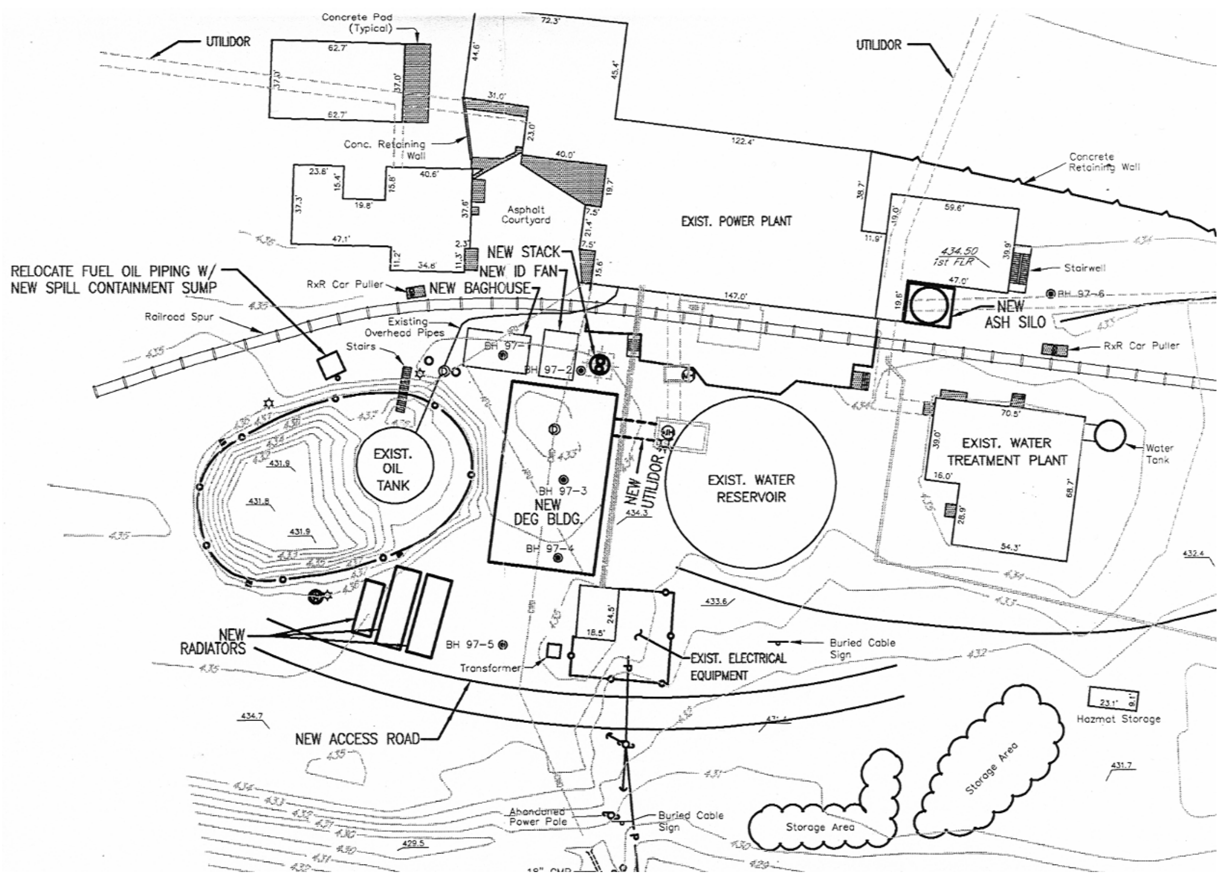


Figure 27. UAF Site Plan Showing Coal-Diesel Installation

Construction of the demonstration plant was initiated on June 15, 1998. Specific activities carried out by month and by year are highlighted below:

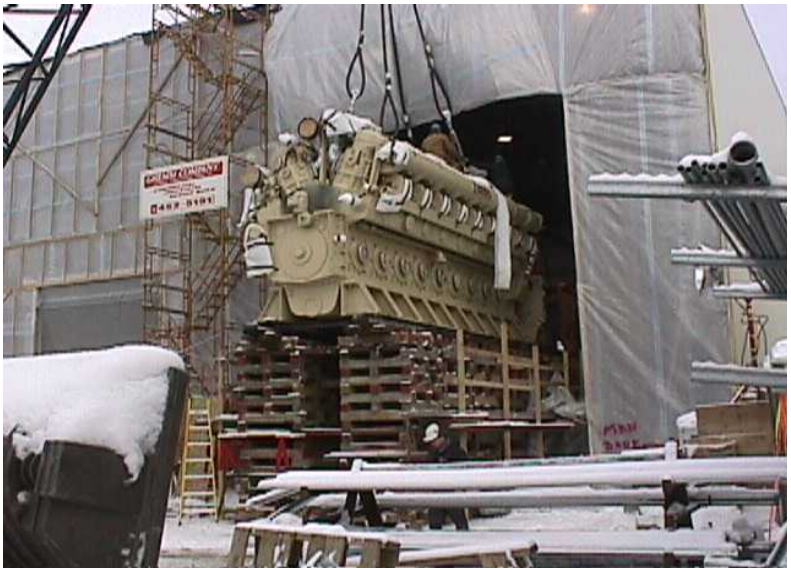


Figure 28. Installation of Coal Diesel Engine at UAF

Figure 29. Installation of Coal Diesel Engine at UAF

June 1998

- Built a temporary road for coal truck access to the power plant
- Set up temporary construction offices
- Initiated pile driving for the diesel building's foundation.

July 1998

- Initiated preparation of foundations for the Coal Diesel engine and the Diesel-Generator building.

August 1998

- Continued work on foundations for the Coal Diesel engine and the Diesel-Generator building
- Initiated relocation of an existing fuel oil tank (to provide room for the DEG building).

September 1998

- Concrete was placed for the Diesel Engine Generator and Heat Recovery Steam Generator foundations.
- The existing fuel oil tank was relocated and construction started on its containment basin.
- Piping in the Utilidor was started.
- Underground piping was completed
- Electrical underground conduit construction was in progress

October 1998

- Installed engine radiator foundations
- Installed temporary and permanent fuel piping to the power plant and fuel storage tank.
- Completed construction of the spill containment basin and truck fill stand for the fuel oil storage tank.
- Installed piping modifications and extensions in the existing plant
- Installed cable tray and prepared for electrical modifications to the existing plant
- Ordered building steel, building siding, station service transformer and medium voltage switchgear (for existing plant tie-in), HVAC equipment and stack.
- The building foundation was completed and is ready for installation of the DEG and building steel
- The 15MMBTU/hr used Babcock and Wilcox boiler to be utilized for the CWF demonstration was purchased and was shipped from Paducah, KY to Portland, OR for repairs

November 1998

- The coal engine's heat recovery steam generator and selective catalytic NOx reduction reactor were ordered from Deltak.
- Aeropulse was selected as the vendor for the coal engine's baghouse.

December 1998

- Erection of the structural steel for the DEG building was completed.
- Electrical cable tray work continued.
- Piping work continued.
- Following successful shop tests at Fairbanks Morse, the demonstration engine was prepared for shipping and left the factory on December 30.

March 1999

- Coal-diesel engine delivered to UAF and installed in new diesel generator building (see Figure 28 and Figure 29 showing the engine installation and Figure 31 showing the building plan view). Figure 30 shows the unloading of the generator in preparation for installation. Figure 32 and Figure 33 show the installed foundation for the diesel engine.

Figure 30. Unloading Generator Stator

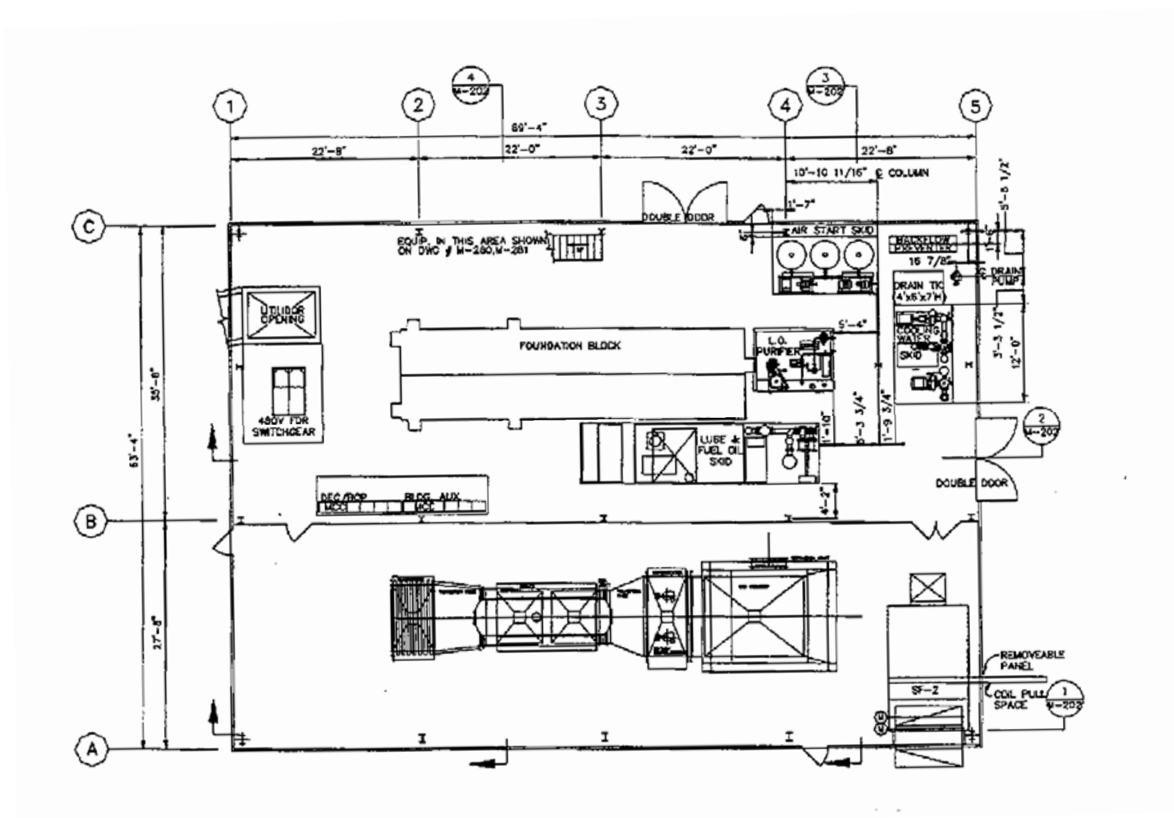


Figure 31. Layout of Coal Diesel Facility

Figure 32. Coal Diesel Engine Mounted on Foundation at UAF

Figure 33. Installed Coal Diesel Engine

April 2000

- Start up of the demonstration plant including engine, generator and heat-recovery steam generator (HRSG). Figure 34 shows the initial start up.

May-August 2000

- Emission Control System components for the demonstration plant including baghouse and SCR Unit were fabricated and installed (see Figure 35 and Figure 36).

Figure 34. Initial Start up of Coal-Diesel Engine Facility at UAF

Figure 35. Baghouse Components Arrive at UAF

Figure 36. SCR Reactor being Manufactured

September 2000

- Acceptance Testing of the diesel engine generator, HRSG, and Selective Catalytic Reduction (SCR) Unit.

April 2001

- Manufacturing of extra sets of hardened components for coal-fueled diesel engine operation (Figure 37 shows components).

May 2001-August 2003

- Continuation of commissioning tests of the 18-cylinder coal-fueled diesel engine. Certain one-time start up problems were observed by UAF which were eventually corrected by Fairbanks Morse. These included (a) the need to increase cooling capacity of the cooling tower in warm weather and (b) oil leakage in the diesel injector fittings.

- Relocation of the demonstration tests to the Beloit, WI facility approved by DOE.

March-June 2004

- Final tests conducted on the 18-cylinder UAF diesel engine facility. The commissioning tests of this facility continued only as long as UAF matching funds allowed.

Figure 37. Engine Block, Cylinder Heads, and Cylinder Liners during Assembly

3. CWF Engine Testing at the Manufacturer

3.1 Summary of Testing

Coal water fuel derived from Alaska coal was successfully used to operate a Fairbanks Morse Model PC-2.6 stationary diesel engine at moderate load (250 hp). This load was equivalent to 25% of full load for this PC-2 engine configuration. To the best of our knowledge, this was the first operation of a diesel engine on coal in the world since the 1996 tests were conducted at Cooper-Bessemer (as part of this same demonstration project). Fairbanks Morse had never before operated an engine on coal, so the transfer of technology from Cooper-Bessemer can be declared as now successful. Appendix C summarizes the earlier test results from the Cooper-Bessemer engine.

On April 14, 2004, a coal water slurry mixture was successfully injected and combusted in a two-cylinder test engine at Fairbanks-Morse Engines Test Facility located in Beloit, WI. EERC prepared the coal-water fuel (CWF) using the hydrothermal process from Alaska sub-bituminous coal (Usibelli mine). The CWF was successfully fired in the engine for close to an hour at 17% load, then the engine was switched back over to DF2. Following the switchover, the engine was again switched over to CWF for another 9 minutes run time. Towards the end of operation, there was a baghouse fire attributed to glowing unburned coal embers in the exhaust, which occasionally occurs in new gasification plants during shakedown testing.

The specially designed injectors functioned flawlessly on CWF. However, combustion repeatability was not as good as DF2 stability, most likely due to the fact that the combustion phasing was not tuned to the optimum point. The injector traces showed that the fuel injected into the cylinder was repeatable and consistent. Cylinder pressure was monitored to reveal rate of heat release, mean effective pressure, and peak pressure. Statistical analysis was used to judge repeatability of the combustion events for several intervals over the duration of the testing. The Coefficient of Variation (COV) of indicated mean effective pressure (IMEP), when operating on CWF, was 5-12%, which means that there was room for improvement. This improvement would come from optimization of the pilot injection timing.

Engine-out NOx was only 150 parts per million at 25% engine load, compared to 430 ppm for diesel fuel operation (60% reduction). The 150 ppm is equivalent to 0.14 g/kWh or 0.45 lb per million BTU. This was an "engine-out" figure and the coal diesel stack emission level of NOx will be 80-85% lower because an SCR unit part of the design. The estimated stack emission at 25% load point would be about 30 ppm, which is equivalent to 0.03 g/kwh or 0.09 lb per million BTU. This is well below the required standards for coal fired power plants.

On selected CWF combustion events where the majority of injected fuel energy was released, the CWF energy release was similar to DF2 energy release (~90%). This result was extremely positive because there was still much optimization to be done.

The burn time of CWF was only 6-7 crank angle degrees longer than that of DF2, which was acceptable for engine efficiency and power level. The ignition delay was close to twice as long when compared to DF2, and this can be addressed by advancing the start of injection. The longer burn time may be attributed to the large amount of pilot fuel interfering with the CWF diffusion flame and consuming available oxygen.

Based on the shape of the heat release curves, CWF combustion exhibited only mixing-controlled combustion, where the DF2 fuel combustion showed an initial pre-mixed combustion followed by mixing-controlled combustion. The pre-mixed combustion with DF2 fuel occurs at a stoichiometric F/A ratio, so the charge temperatures are higher, producing much more NOx.

In future testing it is recommended that the pilot injection be moved so that it ends before or shortly after the main injection event. This will allow better comparison between pure DF2 injection as well as comparison to the previous Cooper-Bessemer engine runs of CWF done in the earlier phases of the project (1996).

3.2 Coal Water Slurry Preparation

The coal water slurry preparation used in the April 2004 engine tests was a North Dakota Energy Research Center-prepared coal slurry mixture used successfully in the previous phases of the project. Table 8 shows the composition of the successful coal slurry fuel formulation.

Table 8. Composition of Coal Water Slurry Mixture

Coal Percentage in slurry (by weight)	47.7 %
Coal Energy Content	10,610 Btu/lb
Stabilizer Content	32 g Xantham Gum
Slurry Density	1.1 g/mL

3.3 Description of the Fairbanks-Morse Engines Test Facility

3.3.1 Standard Two-cylinder Test Engine Description

The two cylinder test engine used for the tests was located Beloit, WI, at the Fairbanks-Morse (FME) Experimental Engine Test Facility. A photograph of the engine test area is shown in Figure 38**.** The test engine is a two cylinder version of the Colt-Pielstick PC2.6 Engine, which in the typical configuration has 18 cylinders, but is also scalable to match FME customer's individual needs. Power produced from the engine is absorbed with an AC dynamometer coupled to a resistive load bank.

The standard configuration for the two cylinder engine is shown in Table 9 when operating on standard diesel fuel. The modifications made to allow the engine to run on CWF are discussed in the following section.

Figure 38. Test Engine Area

Table 9. FME Two-cylinder Engine Specifications

Bore	400 mm (15.74 inches)
Stroke	460 mm (18.11 inches)
Compression Ratio	11.4:1
BMEP	2220 kPa (322 psi)
Power	1050 kW (1400 bhp)
Maximum Cylinder Firing Pressure	14.4 MPa (2100 psi)
Operating Speed	514 RPM

3.3.2 Modifications Made to the Test Engine for CWF Operation

The main modification made to the test apparatus was the fuel delivery system. Pilot injectors were installed into the engine to deliver the initial DF2 pilot into the cylinder. The main injectors specially designed for coal-water slurry were installed with the modified fuel piping system to take slurry fuel from the day tank. For this phase of the testing, hardened parts were not installed into the engine because short test runs to determine combustion quality were conducted. Due to the flow rate limitations of the main injectors on CWF fuel, the two-cylinder engine was derated to 1000 bhp (745 kW). The fueling system has a three-way valve to allow automatic switching from the CWF fuel supply (located on a scale) to the DF2 fuel supply (placed on a separate scale). Once the switch is made, the changeover from one fuel supply to the other is simply the amount of time required to purge the injector supply lines of the other fuel. The switchover (both ways) was observed to occur seamlessly without any particular problems.

The engine was started and stopped on ordinary diesel fuel (DF2); this fuel is used to purge the fuel system of any CWF, as any CWF that remains in the fuel system will harden if not immediately removed before the water evaporates from the slurry. A simplified schematic of the fuel system is shown in Figure 39.

3.3.3 Bench Testing of CWF Injector System

The bench testing noted during this program revealed areas of that would require some minor design changes as well as the need for the proper storage of the CWF. The start of initial bench testing was started with a new batch of CWF which some injector performance problems arose until the proper level of surfactant was determined to have proper injection. During this period some plunger seizures were experience during the switch over from DF2 to CWF. Once the level of surfactant was achieved, all of the prototype injectors were tested and found to be acceptable.

However, the elapsed time between bench tests was significant over the 2002-2004 time period. Once when bench testing of the injectors restarted, after a period of 1.5 years, the success with the injection systems could not be repeated. Various tests were repeated to verify the level of surfactant required, and repeated with a new supply of this material without success. A new batch of fuel was ordered and success was achieved using the original level of surfactant. One very important item learned during this period is how the CWF is stored during extended period of time is very important. It was learned during some investigation that during the 1.5 year that the CWF was stored it had been an unheated build during winter months and probably had froze. This apparently had altered the make up of the CWF not allowing the surfactant to react properly with the fuel.

We also used a matrix to determine the percent solids, type of surfactant and level of surfactant required to have the CWF inject properly and eliminate the possibility of a shuttle seizure during switch over from DF2 to CWF and back to DF2 using CWF produced by CQ and EERC.

We also determined that the internal oiler supply pressure had to be increased from 500 to 1000 psi to assure that the CWF did not migrate into the shuttle piston to bore clearance, as well, in the nozzle body bore to valve clearance while the shuttle was recharging. This was required since it was noted that the CWF supply pressure needed to be raised to approximately 450 psi to recharge the shuttle between injection cycles.

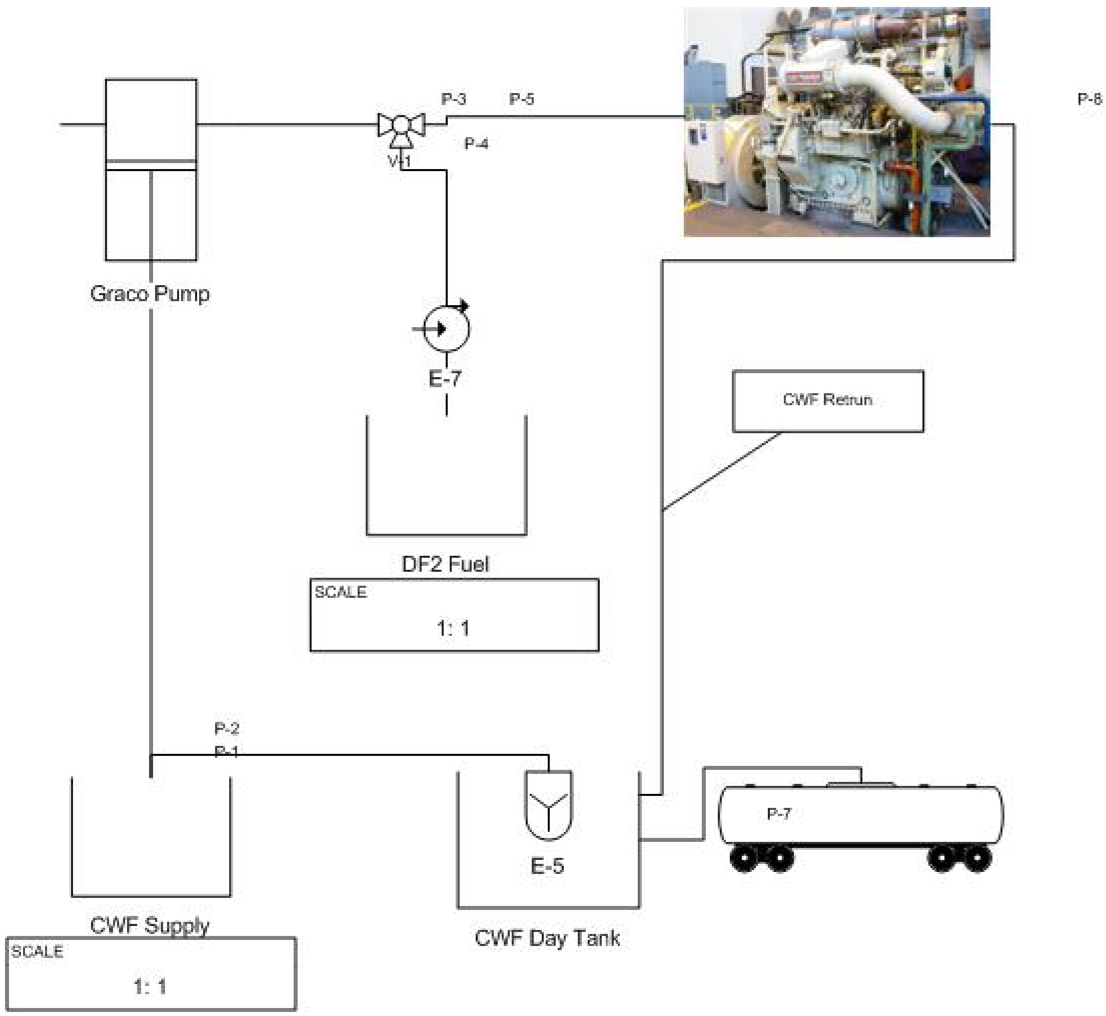


Figure 39. Schematic of Fuel System

3.3.4 Data Acquisition System and Controls System Description

Control of the engine is achieved using a Rockwell Automation programmable logic controller (PLC). To ensure proper engine operation and operator safety, essential parameters such as exhaust temperature, cooling water and lubricant temperature, intake manifold temperature and pressure, engine speed, turbocharger speed, and baghouse pressure drop were all monitored with the PLC. Since these parameters are already monitored for the control of the engine, when designing the data acquisition system, it was decided to port these parameters directly over to the data acquisition computer. Besides these parameters, high speed data acquisition was used to monitor and record the injection and cylinder pressure for the two cylinders and four injectors on the engine. A photograph of the data acquisition system is shown in Figure 40.

Figure 40. Data Acquisition System

3.4 CWF Engine Test Results

As noted above, the engine tests were conducted using coal slurry fuel prepared by EERC (by processing Usibelli coal). This CWF was successfully fired in the engine for approximately one hour at 17% -25% load, then the engine was switched back over to DF2. Following the switchover, the engine was again switched over to CWF for another 9 minutes run time. A summary of key results is given in this section, and more detailed commentary with additional data plots are provided in Appendix A.

3.4.1 Combustion Results based on Cylinder Pressure

The combustion while operating on coal slurry fuel was excellent on most engine cycles, but we observed some variability from cycle-to-cycle. A cycle is an individual cylinder-firing event. Pilot ignition timing was not yet optimized so combustion would be improved once that step was taken. The combustion of coal slurry was over 95% complete based on the amount of net power that was produced. There was no evidence of unburned coal build up in the cylinder volume or on the liner walls. It is instructive to compare the cylinder pressure traces for the nominal diesel fuel operation with the same traces for coal slurry fuel operation. The difference of the pressure traces between the DF2 combustion and the CWF combustion may be seen in Figure 41 vs Figure 42**.**

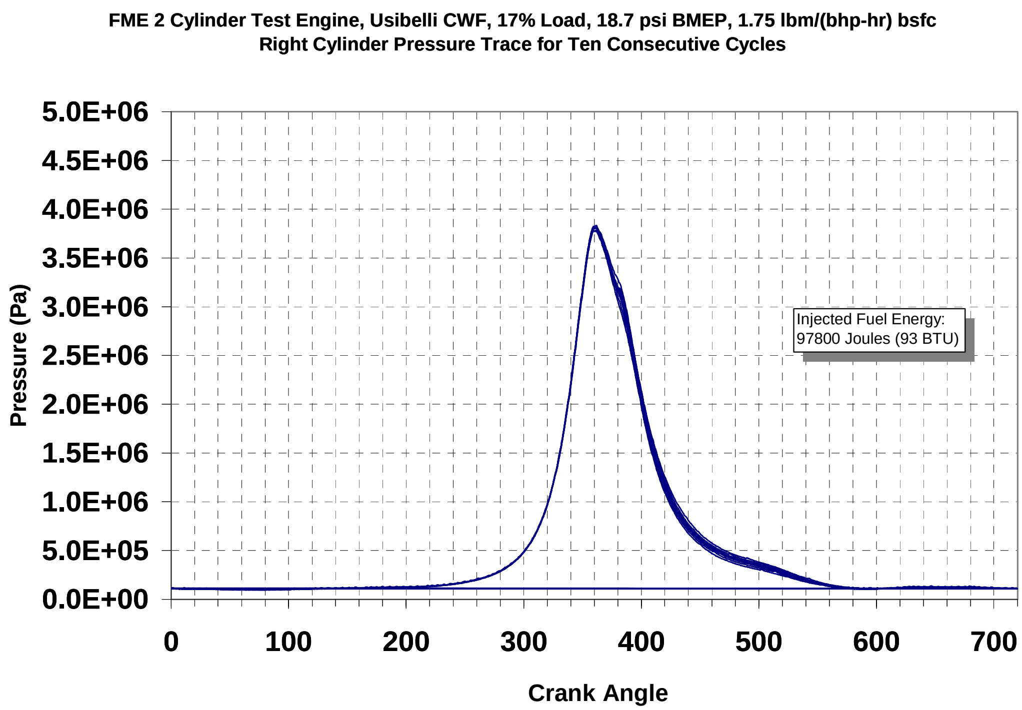


Figure 41. Cylinder Pressure Trace for Coal Water Fuel

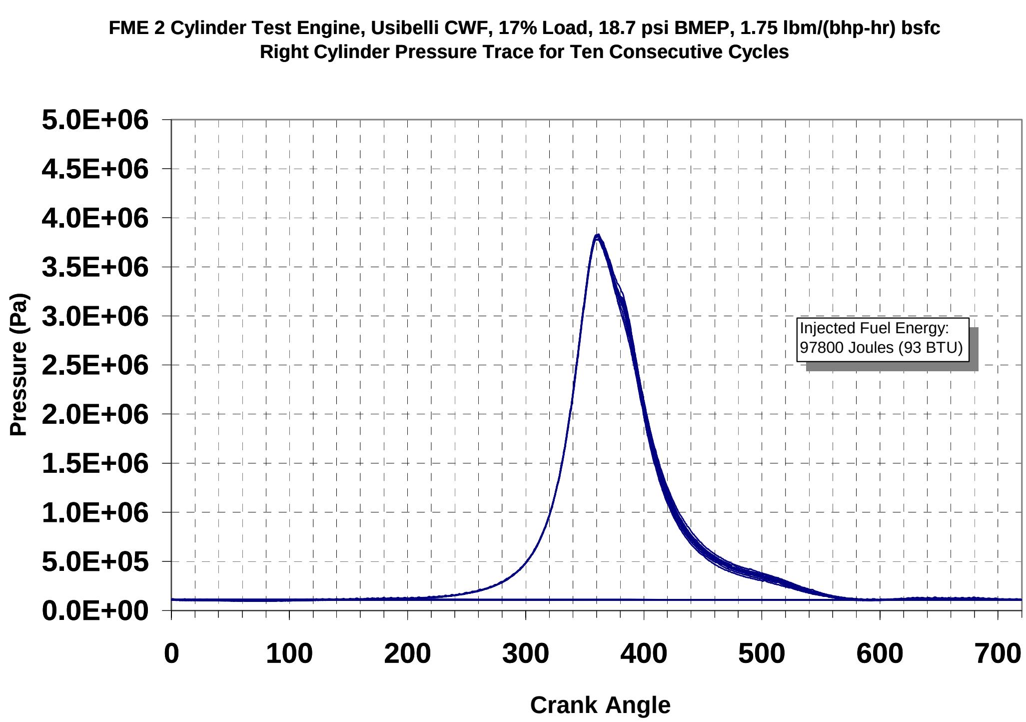


Figure 42. Cylinder Pressure Trace for Diesel Fuel

The DF2 combustion event on average produces a higher peak pressure than that of the CWF combustion. Looking at all of the plots as a package, we observe from the shapes of the pressure curves that the CWF combustion appears to be mixing controlled combustion without the characteristic “spike” near top-dead center indicating premixed burning. We also observe that the DF2 pilot does not end before the main CWF injection event begins. As the CWF is burning, it is competing with the late pilot injection for the same oxygen (in those regions where the sprays overlap). The DF2 combustion event shows an initial energy release from pre-mixed combustion (“spike” in pressure), which is followed by mixing controlled combustion. The slow mixing controlled combustion of the CWF is evidenced by the 6-7 degree longer burn time of the mixture, but the ignition delay of the CWF being twice as long as that of DF2. The DF2 pressure traces presumably had a higher peak pressure due to the fact that the timing of injection was better matched to the burn rates than that of the CWF with the DF2 pilot injection. With tuning this could easily be corrected.

The gross heat release plots showing cumulative energy release are shown in Figure 43 and Figure 44. In Figure 45 we provide a bar chart illustrating the variability in total heat released, cycle by cycle. These figures highlight the difference of the DF2 combustion repeatability versus that of the CWF combustion repeatability. The late burn of CWF on certain cycles also reduced the engine efficiency on CWF, because a percentage of chemical energy was not being converted to work on those cycles. Further engine tuning can correct for this variability.

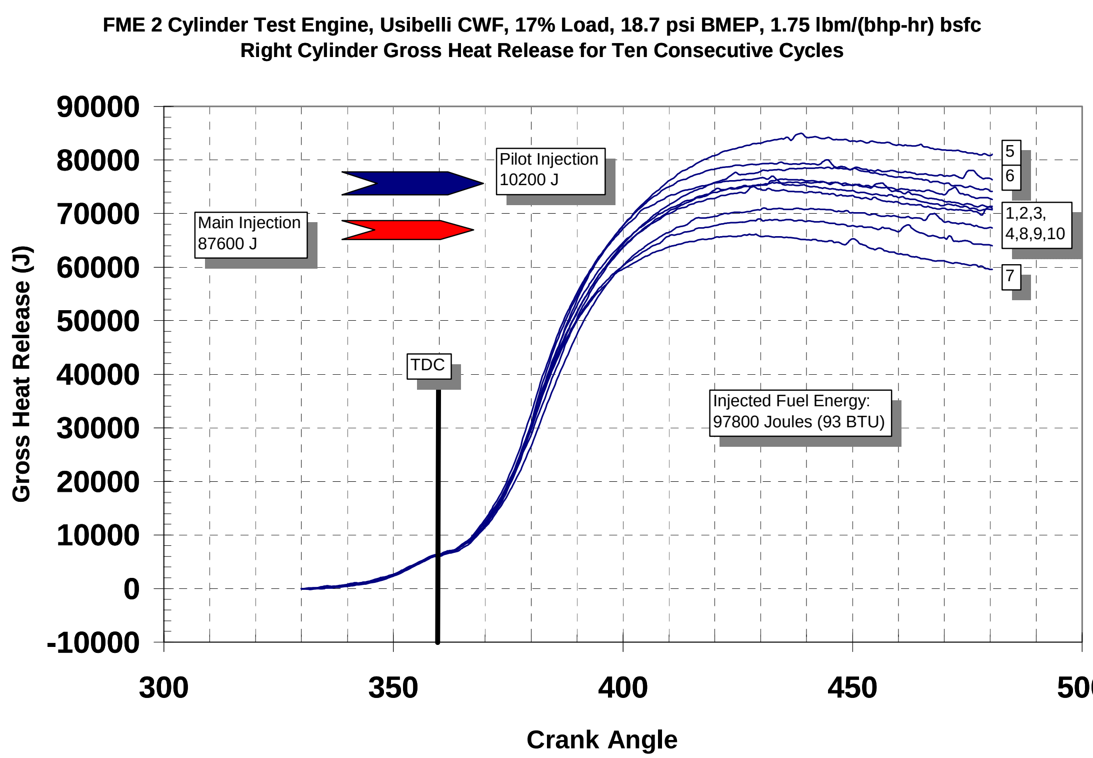


Figure 43. Cumulative Heat Release for Coal Operation

FME Two Cylinder Test Engine, DF2, 25 % Load, 28.2 psi BMEP, 0.681 lbm/(bhp-hr) bsfc Right Cylinder Heat Release for Ten Consecutive Cycles

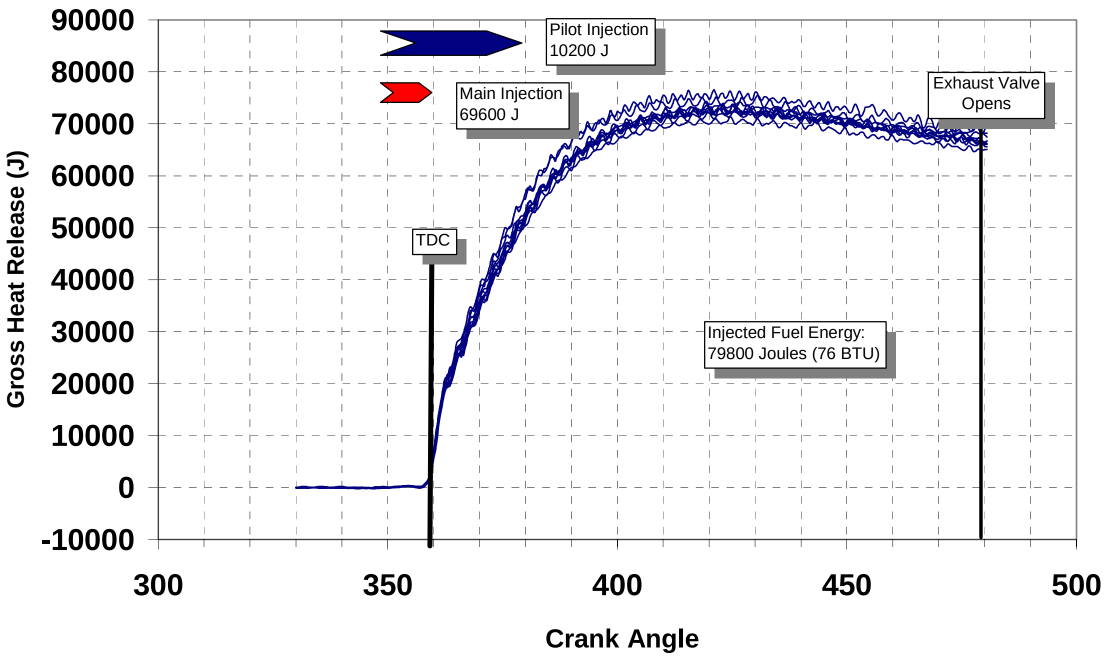


Figure 44. Cumulative Heat Release for Diesel Fuel

FME 2 Cylinder Test Engine, Usibelli CWF, 17% Load, 18.7 psi BMEP, 1.75 lbm/(bhp-hr) bsfc Gross Heat Release Comparison for Ten Consecutive Cycles

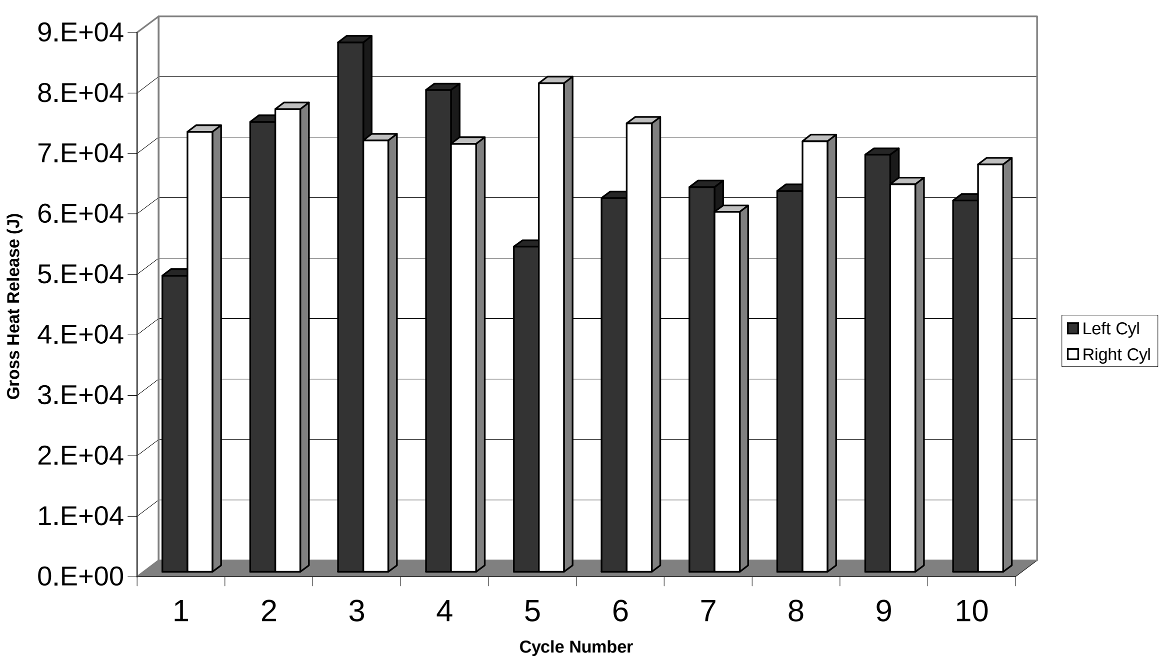


Figure 45. Total Heat Release for Coal on Successive Cycles

The rate-of-heat-release plots, shown in Figure 46 for coal fuel and Figure 47 for diesel fuel, clearly indicate the differences in combustion between the two fuels. The spike seen on the DF2 plots (Figure 47) was a rapid energy release caused by pre-mixed combustion of the DF2 vapor-air premixture, followed by the more steady energy release of the mixing controlled combustion. The CWF rate of energy release plots did not show that initial spike presumably because the pilot injection did not conclude soon enough. There is still good combustion with the CWF, but it is mainly mixing-controlled combustion, which takes longer to initiate and longer to burn. In addition, the CWF is competing with the pilot injection of DF2 for available oxygen. This phenomenon partly explains the higher cycle-to-cycle combustion variability as well as the later burn of the CWF fuel.

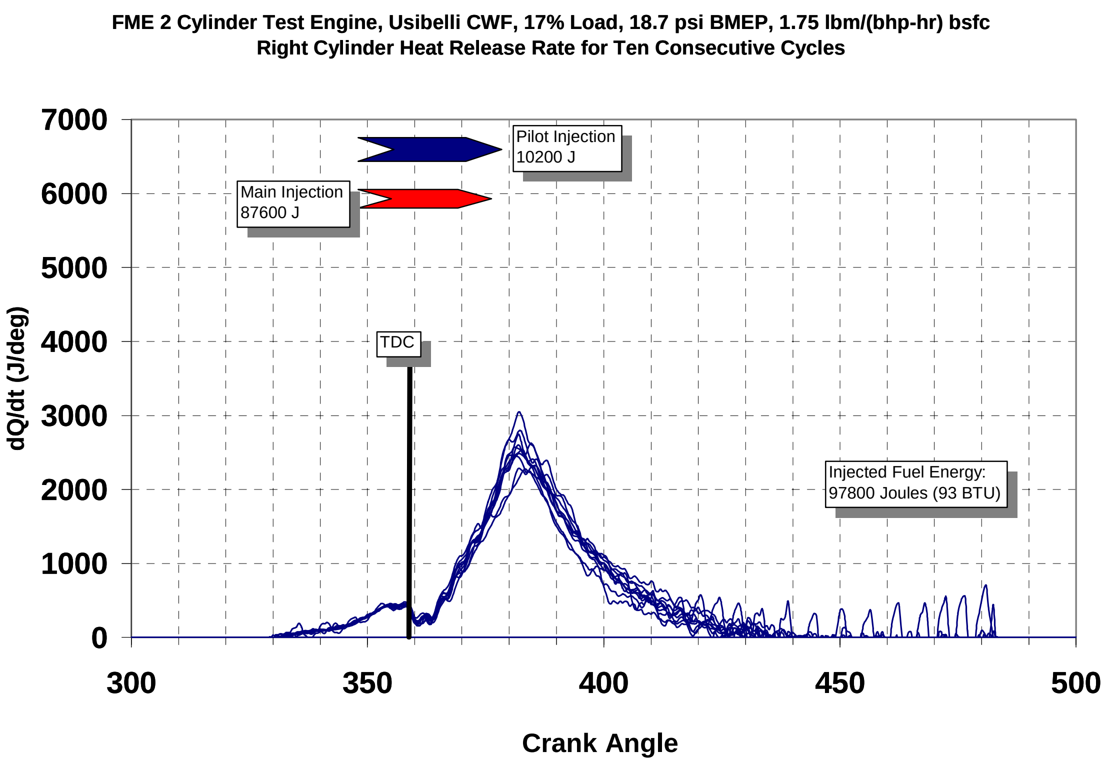


Figure 46. Rate of Heat Release for Coal

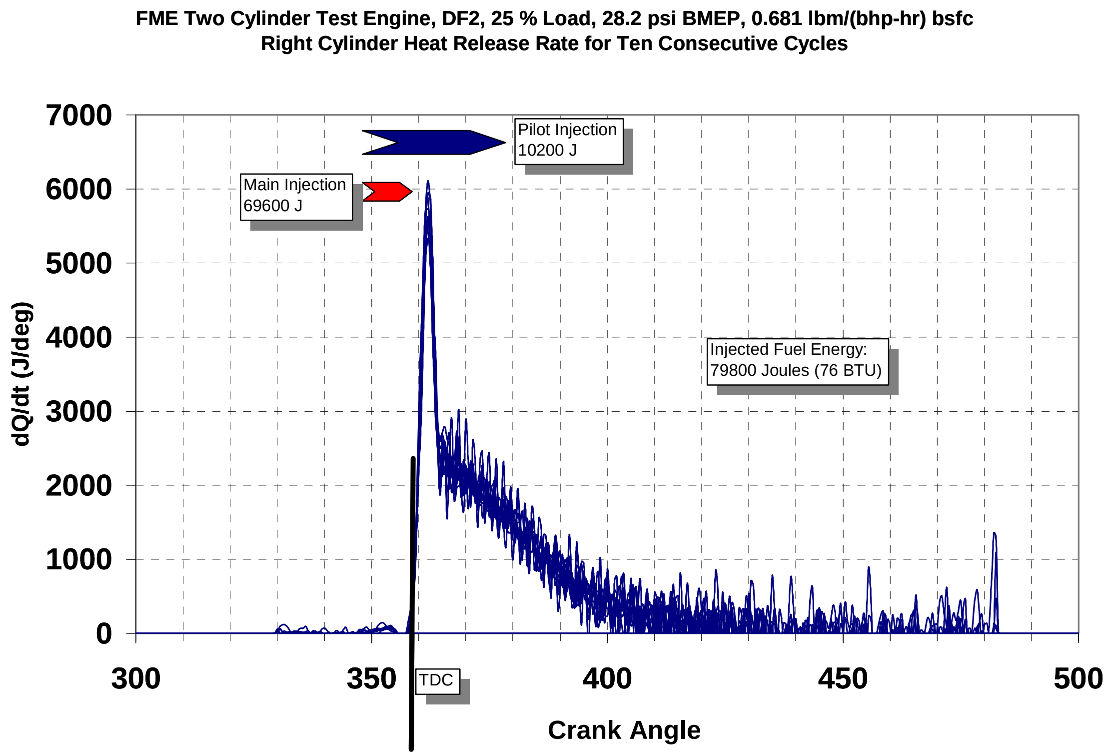


Figure 47. Rate of Heat Release for Diesel Fuel

3.4.2 CWF Injector Performance during Coal Diesel Engine Test

The pilot and main CWF injector performed without any problems during the one hour run on CWF fuel. There were no noted problems experienced during the switch over from DF2 to CWF and back to DF2 at the end of the engine test. Note that preliminary engine testing had been performed on pure DF2 prior to the coal run in order to validate the safety shut down system and switchover procedure.

It should be noted that, based on the success of this engine test, the injector design changes incorporated during the early stages of the bench testing were deemed to be successful.

A typical coal-fuel-operation injector pressure trace, shown on Figure 48**,** shows that both the pilot injectors and the main CWF injectors were able to give repeatable injections for the test, as evidenced by the fact that the pressure traces fall on top of one another. Also from these traces one can see the location (time phasing) of the pilot injection is in relation to the main injection. In future development we recommend moving the pilot injection such that it ends before or shortly after the main injection event begins. This will greatly improve the results. The pilot injection can be seen to have a repeatable ringing after the injection event is completed. From a Fourier analysis of the pressure trace, it appears that there is a pressure wave in the system of around 300 Hz that does not affect engine operation.

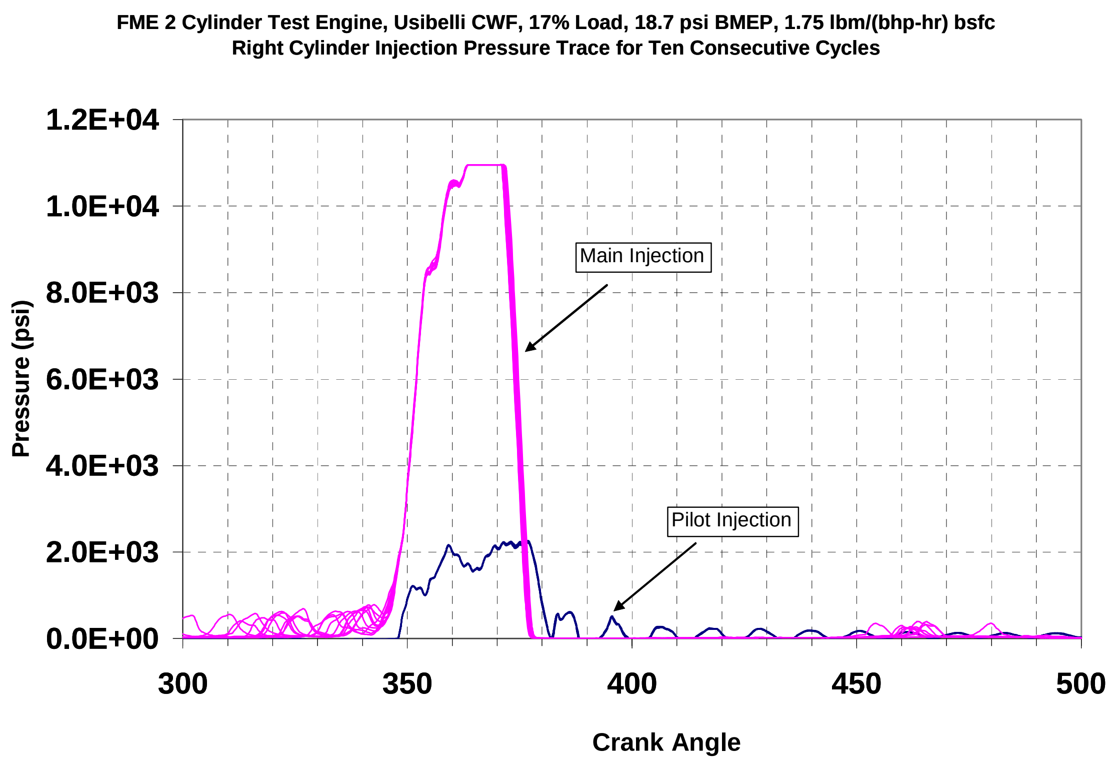


Figure 48. Injection Pressure Trace for Coal

A summary of the operating conditions for the April 14, 2004 inaugural run on CWF are shown in Table 10.

Table 10. Summary of Operating Conditions on CWF for April 14, 2004

Fuel	Diesel	Diesel	CWS	CWS
Engine Load	50%	25%	25%	17%
Engine Power (bhp)	503	254	254	168
Engine Speed (rpm)	506	506	506	506
Pilot %	3.6%	4.7%	11.6%	12.4%
BMEP (psi)	55.8	28.2	28.2	18.7
Fuel Flowrate (lb/hr)	227	173	629	588
Specific Fuel Consumption (Btu/bhp-hr)	8105	12281	10663	15031
Fuel Conversion Efficiency	31%	21%	24%	17%
Air Flowrate (lb/hr)	7532	6232	N/A	5240
Air/Fuel ratio	33.3	36	N/A	18.7
Fuel/Air ratio	0.03	0.028	N/A	0.053
NO (ppm)	1100	430	150	142
CO2 (%)	6.4	5.1	6.3	5.9
CO (ppm)	463	385	>1000	>1000
O2 (%)	11.8	13.9	13.3	13
THC (C3H8) (ppm)	406	N/A	N/A	910

Note that at equal engine power (254 bhp) the specific fuel consumption for CWS was equal to or lower than for diesel fuel operation. It should be remembered that this run was a preliminary result at 25% load without engine “tuning”. Further testing is recommended. A summary of the combustion stability for the April 14, 2004 inaugural run on CWF is shown in Table 11 (note: DF2 numbers are at 25% load, CWF numbers are at 17% load). The variability of coal operation was 7-14% COV compared to 2% COV on diesel fuel.

Table 11. Summary of the Combustion Stability on CWF for April 14, 2004

Variable	Left Cylinder on DF2	Right Cylinder on DF2	Left Cylinder on CWF	Right Cylinder on CWF
IMEP Average (psi)	85	88	74	83
IMEP COV (%)	2%	2%	14%	7%
IMEP LNV (%)	96%	97%	77%	90%

Additional plots supporting the data points shown above are contained in Appendix A.

3.5 Conclusions and Recommendations from Engine Tests

Coal slurry fuel was successfully injected, ignited, and burned in a Fairbanks Morse Model PC2 Engine: Previous to 2004, only Cooper-Bessemer, Sulzer and GE had successfully operated medium and low speed diesel engines on coal slurry fuel. The run was successful in that the injectors worked flawlessly, the expected net power was produced, and the combustion diagnostics showed good combustion (although variable cycle-to-cycle). The injection timing of the CWF could have been further optimized to give even better results, because the ignition delay time of the CWF was very close to twice as long as that of DF2. We attribute this to the pilot fuel being injected after CWF injection, as well as the large amount of pilot fuel.

Injection and Pilot timing: The injector traces showed that the fuel injected into the cylinder was repeatable and consistent. It is recommended that the timing of the pilot injection event be moved and a faster acting cam be installed so that the pilot injection starts before the main injection event and ends shortly after the beginning of the main injection event. Comparing the 2004 run to those done at Cooper-Bessemer in the earlier phases of the program (see page 20), it may be seen that the pilot injection needs to be moved such that it ends before or shortly after the main injection.

Combustion Stability and Repeatability: CWF combustion repeatability was adequate for net power production but not as good as DF2 stability, most likely due to the fact that the combustion phasing was not optimized. Although statistical analysis is typically considered over longer time constants, the fact that COV of IMEP on CWF is 5-12% higher shows room for improvement

Combustion Completeness and Duration: For the more successful CWF combustion events where the majority of injected fuel energy was released, the energy release with CWF was comparable to DF2 energy release (90% to 95%). There was no evidence of build-up of unburned coal in the cylinder.

The burn time of CWF was only 6-7 crank angle degrees longer than that of DF2, whereas the ignition delay was close to twice as long when compared to DF2. The longer burn time may be attributed to the large amount of pilot fuel interfering with the CWF diffusion flame and consuming available O_2.

CWF combustion exhibited what appeared to be mixing-controlled combustion, as expected for a low volatile solid fuel, whereas the DF2 combustion showed an initial pre-mixed combustion followed by mixing-controlled combustion (this is characteristic of diesel combustion). The diesel fuel vapor pre-mixed combustion presumably occurs at or near stoichiometric F/A ratio, where the charge temperatures are at the peak, producing much more NOx during the "spike" in heat release. CWF produces lower NOx not only for this reason, but also because of the water impact on lowering peak temperatures in the flame zones.

Engine-Out Nitrogen Oxides: At given engine load, the engine-out nitrogen oxides (NOx) level was only 150 parts per million at 25% engine load, compared to 430 ppm for diesel fuel operation (60% reduction). The 150 ppm is equivalent to 0.14 g/kWh or 0.45 lb per million BTU. This was an "engine-out" figure and the coal diesel stack emission level of NOx will be 80-85% lower because an SCR unit part of the design. The estimated stack emission at 25% load point would be about 30 ppm, which is equivalent to 0.03 g/kwh or 0.09 lb per million BTU. This is well below the required standards for coal fired power plants.

Need for suppression of "sparklers": The cyclone used successfully in the earlier Cooper engine tests was not installed for these Fairbanks Morse engine tests, allowing any glowing coal embers ("sparklers") to be carried into the exhaust ducting. As a result a fire was inadvertently started in the baghouse and some bags where damaged. The root cause of the fire is an undersized fan and glowing partially burnt carbon rich particulate matter. TIAX and FME developed a cost-effective remedy to this problem, which included fan improvement and unburnt particle suppression.

4. Environmental Performance

4.1 Introduction and Emission Targets

The development of a low-cost emission control system for the coal-fueled diesel engine is necessary for its commercial success in small power generation systems. Because the coal-fueled diesel engine is a new technology, no emission limits have yet been imposed. Regulations are currently applied to all new coal-fired stationary sources larger than 250 MMBtu/hr fuel input, and we expect that regulations in some form will be extended to all new small coal-fired stationary power sources in the near future. When this occurs, the coal-fueled diesel system must be competitive with other technologies for power generation in terms of exhaust emissions. Competing technologies for power generation in the range below 100 MWe include coal-fueled fluidized bed steam plants, as well as internal combustion engines and gas turbines designed to burn natural gas or oil. The EPA has recently announced New Source Performance emission standards (NSPS) for stationary reciprocating engines which, once promulgated, will take effect in 2011. For power generation engines greater than 3000 hp, these standards call for:

- NOx limit of 0.5 g/bhp-hr (0.7 g/kwh)
- Particulate limit of 0.02 g/bhp-hr (0.03 g/kwh)
- Sulfur limit equivalent to 15 ppm in fuel

These standards are lower than the coal-fired boiler standards which are expressed in lb per million BTU. The conversion is that 0.7 g/kWh equates to approximately 0.15 lb per million BTU.

Technical and economic criteria govern the design of an emission control system for the coal-fueled diesel engine. To be competitive, the coal-fueled diesel engine must be as environmentally sound as the other combustion technology alternatives such as small fluidized bed boiler steam turbine plants or natural-gas reciprocating engine generator sets. The current EPA New Source Performance Standards (NSPS) for coal-fired utility boilers can be met by the conventional combustion technologies, and for the purpose of this study, we have elected to design the emission control system to meet these standards. The current NSPS for the major pollutants, NOx, SO2, and particulate matter, are given in Table 12. Both the capital and operating cost of the coal-fueled diesel system must be on par with that of other combustion sources which meet the standards. Also the emission control technologies chosen must be compatible with one another (for system integration) and with the coal-fueled diesel engine operation.

As part of this demonstration project we engaged R.W. Beck to perform an in-depth conceptual design study leading to the engineering design of the coal diesel system installed at the University of Alaska, Fairbanks. The R.W. Beck engineering design included specifications and competitive bids for all major modules that make up the emission control system for the coal-fueled diesel engine. For each of the three major pollutants, control technologies were identified and specified to meet the technical and economic criteria discussed above.

Table 12. New Source Performance Standards for Stationary Sources

Pollutant	*Required Emission Level Coal-Fired Utility Boilers (over 250 MMBtu/hr-input)*	*Proposed Levels for Small Industrial Boilers (30-75 MMBTU/hr-input)*
NO_x	0.6 lb/MMBtu (150 ppm at 15% 0_2)*	1.0 lb/MMBtu
SO_2	1.2lb/MMBtu and 90% reduction, or 70% reduction if less than 0.6 lb/MMBtu is achieved	1.2 lb/MMBtu
Particulate	0.03 lb/MMBtu	0.05 lb/MMBtu

*The sum of NO plus NO_2 is converted from measured concentration to lb/MMBtu by using the molecular weight of NO_2 (as if all NO were converted to NO_2)

4.2 Uncontrolled Engine-out Emissions vs Control Targets

Emission measurements conducted during coal-fueled engine testing at Fairbanks Morse (see Chapter 3) and at Cooper-Bessemer (see Appendix C) combined with coal-water slurry (CWS) properties provided a sound basis for initially defining uncontrolled emission levels from full scale coal-fueled diesel engines. The emissions characteristics of the ECS were designed to be superior to those of larger, advanced, coal-power options. The projected levels and ECS performance targets were as follows:

Particulates: Commercially-viable, engine-grade, CWS is expected to contain 1 to 3 wt% ash (dry basis). Although this is much lower that the parent coal, particulate control devices are still necessary. With the demonstrated high engine combustion efficiency (99 to 99.5% carbon burnout), uncontrolled particulate emissions have been measured at about 1 to 3 lb/MMBtu. Achieving the coal-fired boiler New Source Performance Standard (NSPS) level of 0.05 lb/MMBtu requires a reduction of about 95 to 98%. In addition to air pollution considerations, particulate control is needed to protect the engine turbocharger from potentially severe wear.

SO2: Engine-grade CWF, depending on the source coal has a sulfur content of about 0.2 to 1.5 wt% (dry basis), which yields SO2 levels in the untreated engine exhaust gas of about 70 to 450 ppm at 11% O2 (0.3 to 2.1 lb/MMBtu). Conservatively using the NSPS for coal-fired utility boilers as a guideline, the overall required NSPS reduction for SO2 is currently 90 or 70%, depending on the uncontrolled emission level. Considering the low sulfur content of engine-grade CWF and the relatively small power plant capacity of expected engine installations, 70% reduction of SO2 in the exhaust gas has been chosen as a reasonable target.

NOx: Measured emissions of NOx from the coal-fueled engine are about 30% to 40% of those of conventional diesel engines, due in part to the flame temperature suppression effect of water in the slurry. Measured coal-fueled diesel NOx emission levels of 600 +200 ppm at 11% O2 (1.8 + 0.6 lb/MMBtu) must be significantly reduced to make the coal-fueled diesel engine commercially viable. For example, a reduction of 50 to 75% would be necessary to meet the NSPS coal-fired utility boiler standard of 0.6 lb/MMBtu. However, recognizing that state and local regulations are often more stringent, and that future NSPS may tighten to the level of low-NOx burners (0.3 lb/MMBtu) or the Tier 4 levels for stationary engines over 3000 hp (0.15 lb/million BTU approximately), the NOx control system was designed to achieve 95% recuction (to 0.25 lb/MMBtu). Furthermore, the coal-fueled diesel emission system incorporated Selective Catalytic Reduction (SCR), a control method considered Best Available Control Technology (BACT) by many regulatory agencies.

CO and Unburned Hydrocarbon Emissions: The combustion characteristics of the CWF fuel in both the Fairbanks-Morse and Cooper-Bessemer prototype engines have been excellent, as described above in Chapter 3. Carbon monoxide and unburned hydrocarbon emissions are low, in the ranges of 100-300 ppm and 20-200 ppm, respectively. As a result, control methods for these pollutants are not necessary.

4.3 Emission Control Technologies Designed for Coal Fueled Diesel

4.3.1 Engine Operating Parameters

Since the cost of the emission control system depends on the size of the overall system in the range of 10 to 100 MW power output, we selected 12 MWe (two engines at 6000 kWe each) as a representative configuration at the lower end of the range for costing the emission control components. This engine configuration is considered by Fairbanks Morse as representative of future commercial opportunities. Table 13 details the engine operating parameters that are independent of engine size such as the speed, compression ratio, and brake-specific fuel consumption. Also included in Table 13 are the properties of the coal-water slurry and the expected emissions of the major pollutants. The assumed NOx emission level is based on tests results. The SOx emission level is calculated from a sulfur mass balance, assuming that all the fuel sulfur is converted to SO2. The particulate emission is calculated from the ash content of the coal and from an assumed burnout of 98 percent which is representative of engine test results.

Table 13 presents the anticipated reduction required to meet NSPS for the three major pollutants. The SO2 reduction may be less than 70 percent shown if a credit is allowed for the sulfur removed from the coal during the production of the coal-water slurry.

The cost of emission control equipment relative to the power output of the system (e.g., the cost per kilowatt of electricity generated) decreases as the size of the system increases. The cost estimates are presented for a 12 MWe configuration which would be at the lower end of systems that span the range of interest for small stationary engine power plants. Table 14 details the exhaust flow rates and pollutant emissions in pounds per hour for a typical 12 MWe coal-fueled diesel power plant.

Currently, most diesel-fueled and natural-gas stationary engines are not equipped with the kind of emission control technology that we have assumed will be needed to meet NSPS with the coal-fueled engine. The emission control devices will have to be integrated with the engine system components. A general schematic of the integrated system is illustrated in Figure 49. Emission control equipment downstream of the turbocharger will be common to all engines in the system, i.e., exhaust gases from all engines would be combined after the turbocharger. Approximate temperatures are indicated in the figure, although the exact temperature will depend on the optimum temperature required for the individual emission control subsystems and on the engine exhaust conditions.

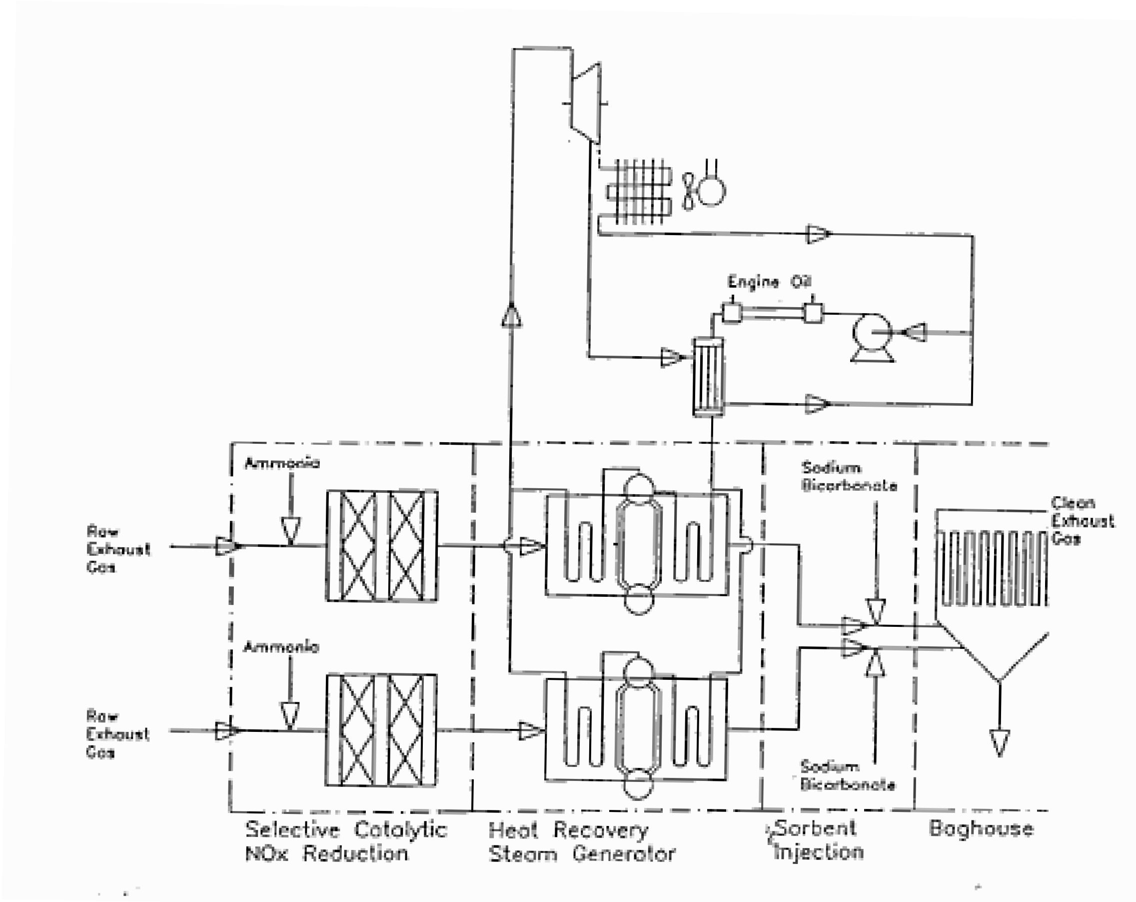


Figure 49. Baseline Emission Control System Configuration

4.3.2 Selection of Emission Control Technologies

The available "aftertreatment" catalyst technology for reducing exhaust gas pollutant levels from gas turbines and steam boilers has rarely been applied to small stationary engines. Most of the experience in removing the major pollutants comes from larger applications such as utility boilers. The emissions from the coal-fueled diesel engine are more closely similar to those from coal-fired utility boilers than to those from stationary engines because of SOx and particulate levels.

Table 7 summarizes the emissions targets (based on utilization of Alaska Coal) and the control methods that will be implemented to reach these levels. The Clean Coal Diesel will include an exhaust gas treatment system, including a cyclone, selective catalytic reduction (SCR), sorbent injection for SO_x control, a baghouse and new exhaust stack to assure appropriate control and dispersion of air emissions. During the prior DOE-METC funded development program, a full scale emission control system was demonstrated to be capable of meeting all of these performance goals.

When operated on coal fuel as designed, the UAF coal-fueled diesel operation was expected to result in significant reductions to overall annual emissions from the UAF power plant. In any extended period of maximum coal-diesel utilization, the four criteria pollutants (SO_2, NOx, particulates and CO) are estimated to be 35-50% lower than emission levels that would be experienced without the coal-diesel in operation. A full air pollution operating permit was applied for and granted to UAF for the operation during the demonstration project.

As part of the demonstration project, an engineering company completed detailed specifications and purchased components under competitive bidding for the coal diesel's emission control system that was installed at UAF. Included in the design document delivered to DOE were conceptual arrangements, heat and material balances, and performance requirements for the silencer, cyclone, SCR reactor, sorbent injection system, and baghouse. This information provided the basis for the A&E team's preliminary and detailed designs.

Table 13. Assumed Engine Operating Conditions and Engine-Out Gases

Coal Properties (assuming bituminous coal after ash cleaning)

Ultimate Analysis:	81.2% C
	5.53% H
	10.07% O
	1.83% N
	0.91% S
	0.38% Ash

Lower Heating Value: 13,736 Btu/lb
Coal-Water Slurry Composition: 53 wt% coal (preliminary spec.)

Engine Operating Parameters (similar for Cooper-Bessemer and Fairbanks Morse)

Compression Ratio:	10.8
BMEP:	200 psi
Power:	419 hp/cylinder
BSFC:	6300 Btu/hp-hr
Speed:	400 rpm

Combustion Conditions

Manifold Air Temperature:	250°F (preliminary)
Exhaust Gas Temperature:	880°F
Stoichiometry:	231% Theoretical Air

Pollutant Levels (Engine-Out):

	Expected Concentration: (preliminary)		Lb/ MMBtu	Assumed Reduction Required
	ppm	mg/ liter		
NOx	600		1.8	87% to meet 0.25
SO2	305		1.3	70%(2)
Solids(1)		1.05	1.5	98%

Notes:
(1) At 98% burnout
(2) Reduction may be less than if a credit for sulfur removal during production of coal-water slurry is allowed.

Table 14. Systems for 12 MWe Coal-Fueled Diesel Operation

Engine	***Engines***
Application	Power
Number of Engines	2
Net Power Output (MW_e)	12
Total Horsepower	16,772
Coal Flow (lb/hr)	7,692
Heat Input (MMBtu/hr)	195.66
Exhaust Gas Flow (scfm)	
Single Engine	23,260
Total (all engines)	46,520
Exhaust Gas Flow (klb/hr)	
Single Engine	103.0
Total (all engines)	206.1
Emissions (lb/hr)	140
SO_2 (0.9% S in coal, 1.3 lb/MMBtu)	
NO_x, as NO_2 (600 ppm or 1.8 LB/MMBtu)	190
Solids (0.4% ash and 98% burnout)	183

4.3.3 Emission Control System

A schematic of the Emission Control System (ECS) which was designed and actually installed and tested for Cooper-Bessemer's 1.8 MWe coal-fueled engine is shown in Figure 50. The system is comprised of the following eight subsystems:

In-cylinder NOx reduction
Cyclone
SCR reactor
Heat exchanger
Sorbent injection, baghouse
Induced draft (ID) fan
Flue gas sample conditioning and analysis

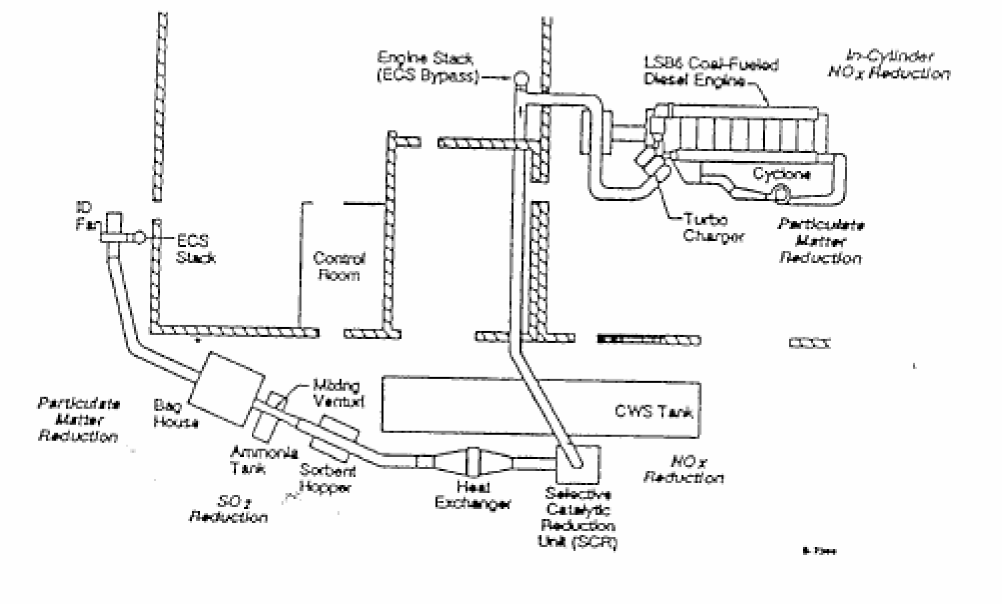


Figure 50. Baseline Emissions Control System Configuration

In operation, exhaust gas from the engine first enters the cyclone where relatively large particulate matter is removed to protect the engine's turbocharger. Gas exiting the cyclone goes to the turbocharger where the temperature and pressure are reduced to approximately 800-850°F and 20 in. w.c., respectively. The first subsystem in the ECS is the SCR reactor where NO_x is reduced by about 80%. Then the gas enters a water-cooled heat exchanger which reduces the gas temperature from 800-850°F to 350°F, simulating a heat recovery steam generator. After the heat exchanger, sorbent is injected into the flue gas in a mixing venture, reducing SO_2 by about 70%. The exhaust gas and sorbent mixture enters the baghouse where the spent sorbent is removed from the flue gas. After the baghouse, the clean exhaust gas flows through the ID fan and to the stack. The ECS control room is located central to the major components of the ECS and contains the flue gas analysis system, data logger and control panels for the ECS subsystems. From this, room operators can control and monitor the performance of all of the subsystems in the ECS. Another engineering plan view of the Emission Control System is provided as Figure 51 below.

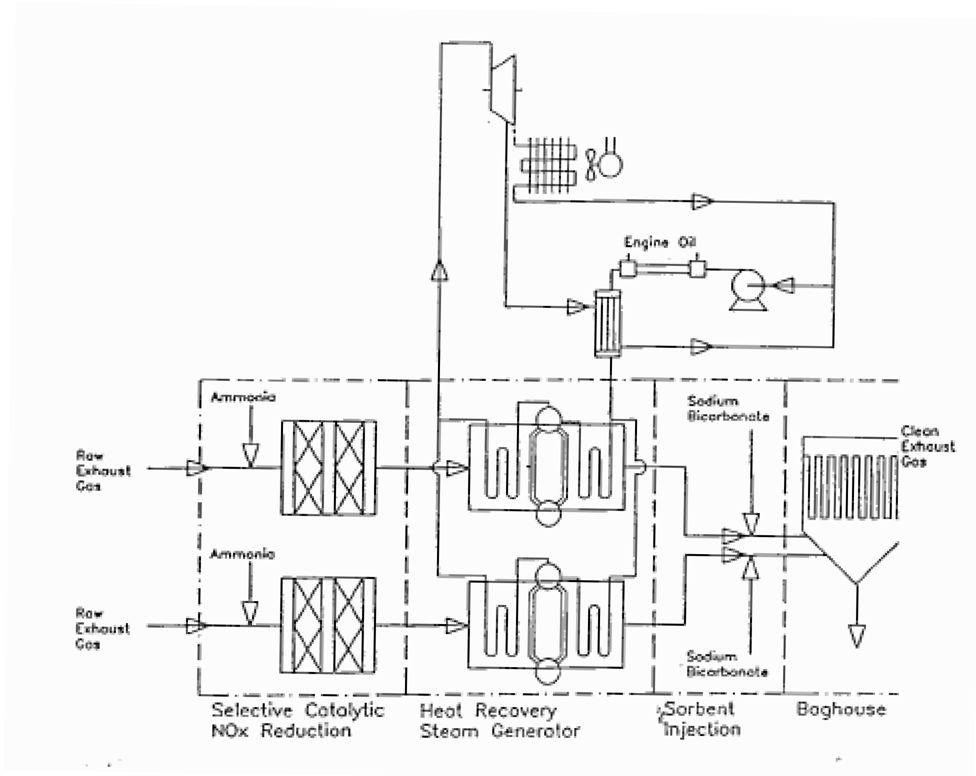


Figure 51. Baseline Emission Control System Configuration

4.3.4 Emission Control System Performance

This section presents the performance of the ECS installed on the prototype Cooper engine in 1996 early in this demonstration project, while processing flue gas from the engine operating on coal water slurry (CWS). A total of about 160 hours of operation were conducted on CWS. While CWS operation occurred in a series of tests, for clarity, this report presents the performance results as a single effort.

Overview of Emission Test Results on Coal (as part of LSC-6 Engine Testing)

The system shakedown was conducted with the engine firing diesel oil. At the end of this shakedown period, the SCR system was achieving up to 90%NOx reduction, and the sorbent injection system achieved up to 80% SO_2 capture. Baghouse performance was not measured during shakedown, and the cyclone was not installed during shakedown.

Initially the SCR was able to achieve up to 90% NO_x reduction. However, performance dropped to a steady value of 75 to 80% NO_x reduction. The sorbent-injection performance improved steadily over the course of testing, achieving up to 95% SO_2 capture. The baghouse was able to capture 99.90 to 99.98% of the incoming particulate matter. The final steady state emissions from the system were as follows:

Stack Emission	ECS Performance vs. Goal (NSPS Std.)
0.35 lb/MBtu NO_x	Actual 90% vs. 80% goal
0.08 lb/MBtu SO_2	Actual 80% vs. 70% goal
0.003 lb/MBtu Particulate Mater	Actual 99.9-99.98% vs. 99.5% goal

Both the sorbent injection and baghouse systems exceeded their performance goals by substantial margins. The SCR system exceeded goals initially; but interactions with effluents from coal combustion and operation below the design temperature resulted in below design performance. SCR performance can be achieved by catalyst reformulation, but this identifies the need to accurately know the engine exhaust temperature over the load range. The cyclone is the only system which did not perform adequately. Unfortunately, the cyclone was unable to capture any significant amount of particulate matter, presumably due to the pulsing nature of the flow. It is apparent that a simple cyclone will not suffice for environmental compliance but should be used to protect the turbocharger.

During the coal-fueled testing, the system was able to meet all of the emission performance goals. Figure 52 shows the NO_x reduction for both the SCR and sorbent-injection systems, and the total NO_x reduction by the system during coal operation. The NO_x reduction by the SCR reactor, which provides the majority of the NO_x reduction, definitely diminished through about 50 h of operation. After 50 h of operation, the performance of the SCR reactor steadied out, just below the performance goals of the system. However, NO_x reduction by the sorbent-injection system was relatively constant during the entire coal-fired operation. Within the operating range of the SCR reactor and sorbent-injection systems, the total NO_x reduction was greater than the goal of 80% for all but two test points.

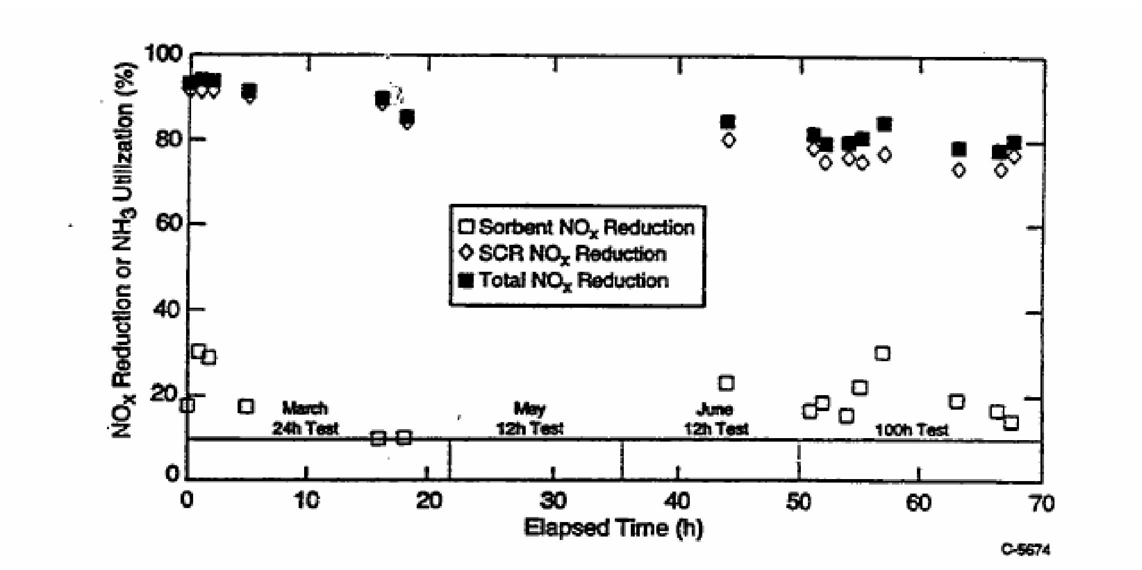


Figure 52. Emission Control System NOx Reduction during Coal Operation

Figure 53 shows both NOx and SO_2 reduction from the system over time for the points where both the SCR and sorbent-injection systems were within their operating limits: SCR inlet temperature 800 - 900°F and sorbent-injection temperature 370 – 450°F. NO_x reduction started high and gradually fell to just about the lower limit of acceptable operation (80% NO_x reduction). For the design conditions SO_2 reduction started just below the lower limit (70% SO_2 reduction) and improved to over 90% SO_2 reduction during the course of operation.

During the majority of testing, the engine was run at 175 BMEP. At this load, the heat input was about 15.2 MBtu/h of CWS and 1 MBtu/h of diesel oil for the pilot. The total system NO_x reduction of 79 – 85% NO_x (82% avg.) reduction translates to NO_x emissions of 0.35

lb/MBtu which compares favorably to current pulverized coal technology. Two tests were conducted early in the test series in which the SCR reactor achieved 90 and 91% NO_x reduction. For these two tests, the total system NOx reductions were 92 and 94% resulting in NO_x emission of about 0.14 lb/MBtu, which is below the 2011 EPA standard for stationary engines over 3000 hp (0.15 lb/MBtu).

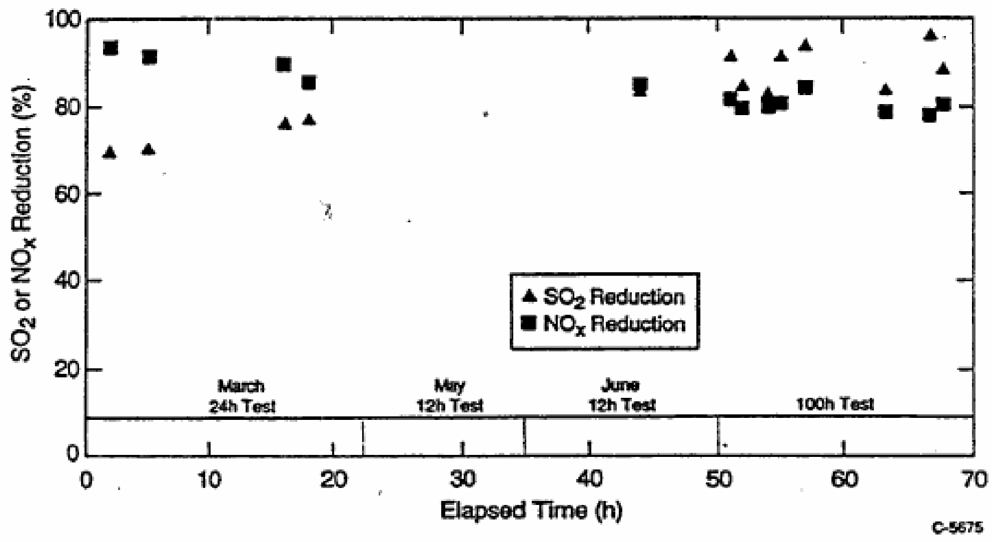


Figure 53. Overall Emission Control System Performance During Coal Operation

The SO_2 reduction for the 175 BMEP tests at design conditions ranged from 70% initially, to between 85 and 95% during the 100-hour test. The initial SO_2 emissions of 0.3 lb/MBtu dropped to a steady state level of 0.08 lb/MBtu. This level of SO_2 emissions was more than an order of magnitude below the 1990 CAA, Title IV requirement of 1.2 lb/MBtu.

Particulate emissions ranged from 6.5E-4 lb/MBtu during the first 24-hour test to 2.3E-3 and 3.0E-3 lb/MBtu during the 100-hour test. These levels of emissions were about one order of magnitude below the 1990 CAA goals of 0.05 lb/MBtu.

As seen, the performance of the emission control system (ECS) met or exceeded all expectations. When the technology is commercialized, this design and associated equipment can be utilized to keep the surrounding environment free of significant releases of pollutants.

5. Cost

5.1 Summary

The levelized cost of electricity for the owner-operator of a coal diesel plant depends on several factors such as amortized installed capital cost, annual hours of operation, coal slurry fuel price, emission control costs, maintenance and parts replacement costs, system efficiency etc. The framework for analyzing the coal diesel cost of electricity originally was formulated by Arthur D. Little, Inc. under contract to DOE in 1986 (see reference 40 "Coal Fueled Diesel Systems for Stationary Power Applications, Task 1 – Topical Report, Assessment of Merit," DOE/MC/22181-2242). This analysis has been updated with revised assumptions several times since, as new information has come to light on costs of processed coal slurry fuels, emission control, engine component replacement intervals, engine efficiency, and other factors. Subsequent testing of both the Cooper-Bessemer prototype engine and the Fairbanks Morse test engine on coal fuel provided data to enable us to revise certain key assumptions in the economic model (e.g., especially engine efficiency and injector replacement intervals). The estimated cost of electricity for the coal diesel has changed significantly since 1986 in five areas:

- Coal processing cost estimates
- Cost of delivered and mine-mouth premium raw coal feedstock coals (increased)
- Cost estimates for emission control modules (SCR, spray dryer, etc.)
- Engine efficiency estimates (estimated at 40-42% for first generation and projected at up to 48% for fully mature engine designs).
- Replacement intervals and costs for injector tips, piston rings, and turbine rotors

Table 21 summarizes the current revised cost of electricity estimates. Our conclusions, as explained in more detail below, are that

(1**)** the projected year-2015 cost of coal diesel power is 7.6¢/kWh, (2) economic viability hinges on the cost of CWS production (engine-grade slurry price must be kept below about $3.50/million Btu), and (3) emission control costs represent an equally important cost parameter; they could "make or break" the economic success. As shown in Table 21, the emissions control cost at $2.50 per million Btu of fuel heating value is converging toward the CWS fuel cost. This economic analysis provides perspective on the conditions under which the coal diesel could be cost competitive with natural gas fueled stationary engines.

5.2 Engine Component Cost Premiums Associated with Coal

The use of beneficiated coal slurry necessitates certain modifications to the standard large diesel engine, both in terms of special hardened components, special maintenance practices and more frequent replacements. Although the precise nature of these modifications cannot be determined until further demonstration and durability testing is complete, it is still useful to project approximate cost premiums based on the engine manufacturers' best judgment at this time. Table 15 lists the engine components expected to be affected by coal fuel; these are generally “moving parts” which are exposed to either the fuel or the products of combustion. The estimated cost premium associated with the modification of these components is $1.67 million. That is, a 6000 kW coal fueled engine is projected to cost $5.2 million, on an installed basis.

The cost premiums associated with the critical engine parts are estimated in Table 15. Note that the total premium is $1.1 million at the factory, and that the most expensive items, in terms of adapting the engine to coal fuel, are the turbocharger, the fuel pumps, the injectors and supply system, and the instrumentation. Installation costs will carry a premium for the coal diesel as well, because of extra slurry pumps, tanks and piping; the offset turbocharger and related exhaust manifolding; the extra instrumentation; the diesel pilot system, etc. Permits and compliance testing will also be more costly. Following common cost estimation practice, this is accounted for in Table 15 by applying 50% installation cost to the entire base engine hardware cost.

Table 15. Estimate Cost Premiums for Coal Diesel Engine Components *(20 cylinder, 6 MW engine)*

Engine Components	***Conventional NG or Diesel (1)***	***Coal Diesel***	***Premium for Coal***
Lube System	$30,000	$65,000	
Pistons and Piston Rings	50,000	70,000	
Cylinder liners	100,000	200,000	
Cylinder Head(s)	200,000	300,000	
Valves/Seats	30,000	40,000	
Fuel Pumps, Injectors, Supply System	100,000	300,000	
Instruments	50,000	200,000	
Turbocharger	200,000	600,000	
Cam Shaft and Bearings	50,000	100,000	
Miscellaneous	200,000	250,000	
Subtotal Critical Parts	$1,010,000	$2,125,000	**$1,115,000**
Non-Critical Parts	1,350,000	1,350,000	
Total Engine Cost	$2,360,000	$3,475,000	
Installation Costs (50%)	1,180,000	1,737,500	
Installed Engine Cost	$3,540,000	$5,212,500	**$1,672,500**

(1) Component costs for the conventional engine are nominal representative values based on our broad experience, and do not necessarily correspond to prices of actual current engine parts of any specific engine model.

5.3 Overhaul and Parts Replacement Cost Estimates

The overhaul and maintenance for a coal-fueled engine are significantly greater than for a conventional diesel or natural gas engine. Not only are the parts replacement intervals shorter with coal, but the parts themselves are more costly. Four basic maintenance and overhaul service functions were considered:

- Injector tip inspection and replacement
- Inspection of filters, valve clearance, etc.
- “top-end” overhaul (cylinder head rebuilds)
- “bottom-end” overhaul (pistons, liners, crankcase bearings)

Table 16 shows the intervals and cost estimates for the coal diesel versus the conventional engine. The period analyzed was 20 years; this is sufficient to allow bottom-end overhaul costs to enter the calculation. The total overhaul costs for the coal diesel over 20 years amounts to $3.9 million versus $1.4 million for the conventional diesel or natural gas engine. The injectors are a significant factor (over 25%) in this total; we have assumed $3200 per set of twenty injector tips ($160 per injector tip) and a 500 hour replacement interval. Each of these estimated numbers has a significant uncertainty at this stage of the technology.

On a per-kWh basis, this capital cost translates to 1.21 cents per kWh vs. 0.81 cents per kWh for the standard oil or gas engine, based on 20-year life and 80% load factor. This premium of 1.21-0.81=0.40 cents per kWh is exclusive of emission control equipment costs, which are accounted for separately below.

The present maintenance practices for a 6 megawatt engine are compared to the expected coal diesel practices (as revised in view of engine test results) in Table 17.

The cost associated with this increased level of maintenance and overhaul is projected to be $3.9 million over a 20-year period, vs. $1.4 million for a standard 6 MW diesel or natural gas engine, which translates to $.0052/kWh for the coal engine vs. $.0019/kWh for the standard. It is recognized that injector life would be limited on coal fuel (500 hr is the assumed injector life). The projected interval between major overhauls was also reduced to 25000 hr. See Table 17 for detailed maintenance costs.

In summary, the total amortized cost for purchasing, installing and maintaining a 6 megawatt coal diesel system (exclusive of fuel and emissions control) is expected to be $.0163/kWh, compared to $.0100/kWh for an oil-fired diesel of comparable size, including both capital cost and maintenance. Since the standard busbar cost of producing power with natural gas can range from about 6 to 8¢/kWh, depending on prevailing fuel prices, this premium of $.0063/kWh for a coal-tolerant engine would represent about a 10% increase in power cost, according to our analysis. This premium is by no means prohibitive. In fact, if emission control costs or fuel costs could be reduced significantly by the use of more expensive engine parts or more frequent maintenance, this could be easily justified according to the economics.

Table 16. Engine Maintenance and Overhaul Costs

Description	*Unit of Measure*	*Coal Diesel*	*Conventional Diesel or NG*
Maintenance Intervals			
Injectors	Hours/Cycle	500	2,000
Minor	Hours/Cycle	4,000	8.000
Top End	Hours/Cycle	12,000	25,000
Bottom End	Hours/Cycle	25,000	100,000
Cost Per Servicing			
Injectors	$$$	$3,200	Negl.
Minor	$$$	$10,000	$10,000
Top End	$$$	$60,000	$90,000
Bottom End	$$$	$290,000	$290,000
Servicing Through 1st Rebuild			
Injectors	# of Cycles	50	N/A
Minor (excl. of injectors)	# of Cycles	6	13
Top End	# of Cycles	2	4
Bottom End	# of Cycles	1	1
Cost Through 1st Rebuild			
Injectors	$$$	$160,000	Negl.
Minor	$$$	$60,000 (for 6)	$130,000 (for 13)
Top End	$$$	$120,000 (for 2)	$360,000 (for 4)
Bottom End	$$$	$290,000	$290,000
Total Through 1st Rebuild	***$$$***	***$630,000***	***$789,000***
Allocated Maintenance %			
Injectors	%	25.50%	
Minor	%	9.52%	16.67%
Top End	%	19.05%	46.15%
Bottom End	%	46.03%	37.18%
20-Year Service Cycles			
Injectors	# of Cycles	315	
Minor	# of Cycles	39	20
Top End	# of Cycles	13	7
Bottom End	# of Cycles	6	2
20 Year Cost (Six Major Rebuilds)			
Injectors	$$$/20 yr	$1,008,000	
Minor	$$$/20 yr	$390,000	$200,000
Top End	$$$/20 yr	$780,000	$630,000
Bottom End	$$$/20 yr	$1,740,000	$580,000
Total Maintenance Cost (20 yr)		***$3,918,000***	***$1,410,000***
Total Maintenance Cost (per kWh)		***$0.0052***	***$0.0019***

Table 17. Impact of Coal Fuel on Maintenance and Overhaul Practices

Item	*Standard 6 MW Diesel*	*Coal Diesel Projections*	*20-Year Overhaul Costs**
Injectors	2,000 hr.	500 hr.	$1,008,000
Minor Maintenance Checks	8000 hr.	4,000 hr.	390,000
Top-End Overhaul	25,000 hr.	12,000 hr.	780,000
Major Overhaul	100,000 hr.	25,000 hr.	1,740,000
Total	–	–	**$3,918,000**

* estimate for coal diesel

5.4 Cost to Produce Engine-Grade Coal Fuels

An essential ingredient in the future of coal-fueled diesels is the eventual emergence of a price advantage of the engine-grade coal fuel. Recognizing that (a) fuel oil and natural gas prices will almost certainly continue to rise in the 2015-2030 timeframe, and (b) the extent and timing of the natural gas price increase is virtually unpredictable, we focused our attention on the coal slurry fuel cost and how it might be reduced as much as possible. Our findings are based on detailed inputs from CQ Inc., NDEERC, AMAX and other leading manufacturers of coal slurries. Key parameters were as follows:

- Physical cleaning (resulting in a 1.0-3.0% ash product) is assumed to be sufficient for the CWS to be compatible with the Fairbanks-Morse or Cooper-Bessemer coal diesel engine. This eliminates expensive chemical cleaning steps.

- Dedicated engine-grade slurry facility; plant incorporates cogeneration using reject char and oversize coal.

- Plant capacity: 1.8 million tons per year (supports several power plants in a region comprising 100 engines at an average 5 MW, operating at 80% load factor)

- Capital cost of plant: $78 million

- Typical coal price: $1.66 per million Btu, delivered to slurry plant (includes $.57/MMBtu for transportation cost ($0.275/ton-mile for 590 miles) plus $30/ton for minemouth coal).

- Electricity cost: $0.04 kWh, 175 kWh/ton.

- Total operating cost of plant: $0.57/MMBtu (excludes amortized capital cost)

- Capital recovery costs: 10% interest/12 year payback.

Assume process development costs are already paid off, so that the plant costs are for the "nth plant." Here it is assumed that the R&D investment in coal cleaning technology by slurry suppliers will be fully recovered by the time the first two or three plants reach full capacity.

The results of this analysis indicate that engine grade slurry will cost about $3.00 per million Btu before delivery charges (which is about $1.34/MMBtu more that the delivered source coal). “Low” and “high” estimates for the CWS cost are $2.50 and $3.70 per million Btu (these estimates correspond to 50% less and 100% greater overall coal cleaning cost).

Table 18 shows the projected engine grade CWS price assuming a multi-product plant approach. This analysis shows that slurry can be delivered to the engine plant at under $3.50 per million Btu. Operating and maintenance cost breakdown for the slurry production is shown in Table 19. Over 40% of the cost of the slurry goes to the additives. Much of the remaining costs (electricity, grinding media, maintenance) is associated with fine coal grinding. Capital cost assumptions are for a 250 tons per hour coal cleaning plant addition plus a coal slurrying plant at 100 tph. The coal slurrying plant is far more expensive than the coal cleaning facility.

Table 18. Projected CWS Price for Regional Production

Cost Component	Cost ($/MMBtu)	% of Total CWS Cost
Feed Coal delivered[1]	1.665	47.8%
Coal Cleaning		
O&M	0.188	5.4
Capital Recovery[2]	0.112	3.2
Subtotal	0.301	8.6 %
Coal Slurrying		
O&M	0.513	14.7
Capital Recovery[2]	0.542	15.6
Subtotal	1.055	30.3 %
CWS Transportation	0.461	13.2%
Total	**3.482**	**100.0**

[1]Pre-cleaned feed coal is $45/ton delivered in year 2015 per DOE Report EIA-0383 (2007).
[2]Capital recovery includes 12.5% cost of money.

Table 19. Operating and Maintenance Cost Breakdown for CWS Production

Cost Component	Cost ($/ton Coal feed)	% of Total O&M Cost
Labor	3.39	23.2
Electricity (78.8 kWh/ton)	3.15	21.6
Dispersant	4.91	33.7
Stabilizer	1.28	8.8
General Maintenance	0.81	5.6
Grinding Media and Liners	1.05	7.2
Total	**14.59**	**100.0**

There are two areas where further process developments could reduce the cost of engine-grade slurries: First, the additives and reagents which currently account for about 30% of the slurry cost have not been optimized. These additives and reagents are needed to keep the coal particles in suspension and to tailor viscosity. Second, further effort is needed to identify lower cost slurry formulations which are compatible with the engine.

5.5 Cost of Emission Control System

Based on detailed design and cost estimates based on vendor quotes for the UAF coal diesel installation, we selected one preferred emission control design package for the overall plant economics. Table 20 summarizes this package of emission control modules and gives the corresponding total capital and operating costs. The capital cost corresponds to \$210/kW to \$270/kW depending on power plant capacity. This represents an incremental premium of approximately 20-25% over the basic installed capital cost of the diesel generator set.

Table 20. Emissions Control Subsystem Costs

Power Plant Capacity		***4x6 Cyl 7.2 MW***	***2x20 Cyl 12 MW***	***4x20 Cyl 24 MW***
Emission Control Modules:				
SCR for NOx	Capital $MM	1.957	2.866	5.114
Sorbent injection for SOx	Oper. $/kWh	0.018	0.015	0.014
Baghouse and cyclone for particulate matter.	Total $/kWh	0.024	0.021	0.019

5.6 Levelized Cost of Electricity for Coal-Fueled Diesel Power

Based on the individual cost premiums for installing and operating a diesel engine designed for coal fuel, it is possible to project the total cost of power and determine the relative importance of each economic parameter. Our key findings from this analysis can be presented in the form of Table 21, which lists the elements which contribute to the cost of power. Note that the coal engine designer can afford to consider more expensive components if that results in an engine which tolerates lower cost coal fuel. Also note that the coal-fueled engine is projected to be competitive with the standard diesel at \$11.80/MMBtu distillate fuel oil and \$2.39/MMBtu delivered coal prices that DOE forecasts for the year 2015.

Table 21 illustrates the important cost elements of a stationary engine power plant of 10 MW assumed size, operating at 4000 equivalent full load hours per year (intermediate load factor)

Table 21. Cost of Electricity for 10 MW Engine-Based Power Plant

Cost Element	Coal Diesel	Conventional Diesel Engine	Conventional Natural Gas Engine
Capital cost including emission control (total power plant and building) Amortized capital (20 yrs, 6%)	$1500/kW or $15 million $1,340,000/year	$1000/kW or $10 million $900,000/year	$800/kW or $8 million $720,000/year
Annual power generated (**4000 full load hours**) Heat rate (assumes efficiency 38%) Fuel usage Delivered fuel cost (CWF @ $3.70/MMBtu)	40×10^6 kWh 9000 Btu/kWh $360,000 \times 10^6$Btu $1,330,000/year	40×10^6 kWh 9000 Btu/kWh $360,000 \times 10^6$ Btu $4,240,000/year	40×10^6 kWh 9000 Btu/kWh $360,000 \times 10^6$ Btu $2,040,000/year
Annual maintenance parts and labor (CWF diesel engine requires parts more frequently)	$.009/kWh $360,000/year	$.005/kWh $200,000/year	$.004/kWh $160,000/year
Total annual costs	$3,030,000 7.6¢/kWh	$5,340,000 13.3¢/kWh	$2,920,000 7.3¢/kWh
Annual Savings	$1,690,000 vs oil Approx same as gas		

Notes: (1) All industrial fuel prices are taken from DOE forecast- Report EIA-0383(2007)-for year 2015 (in 2005 dollars). (2) Assumed CWS cost at the coal process plant is $3.20/MMBtu including $2.40/MMBtu raw coal delivered ($47.45 per ton) and $0.80 for processing; CWF transportation cost is typically an additional $0.50/MMBtu. (3) Diesel fuel oil delivered to the power plant is assumed to be $11.80 per MMBtu ($1.60 per gallon). (4) Natural gas is projected to be priced at $5.65/MMBtu.

It is clear from our analysis (see Table 21) that the most important parameters are the cost to clean the coal fuel and the emissions control, which essentially dominate the economics. If the cost of delivered engine-grade slurry can be brought below $3.50/million Btu or more than $2 per MMBtu below the equivalent industrial natural gas price, then the coal diesel can probably be competitive with natural gas by the 2015-2030 timeframe, depending on long-range natural gas price trends.

6. Commercialization Plan

6.1 Addressable Market and Commercialization Sequence

The commercialization path following this Clean Coal Diesel Demonstration and additional needed durability run time by an engine manufacturer is described schematically (left to right) in Figure 54. As a result of this demonstration, when fuel prices are sufficiently attractive, we can assume that Fairbanks Morse, GE, Caterpillar, or some other engine manufacturer will elect to adopt the coal diesel technology and further demonstrates long term durability and availability. At that point both the engine manufacturer and the coal slurry processor (for example CQ Inc.) will have all the know-how and operating data needed to proceed with scale up.

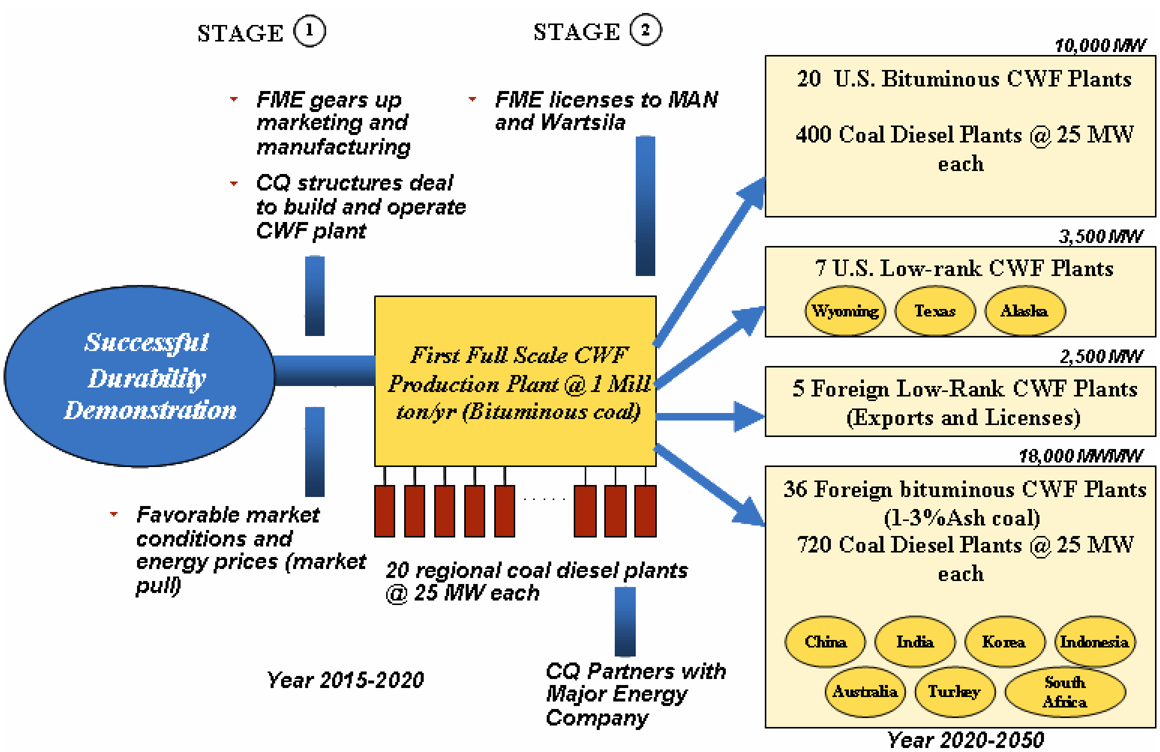
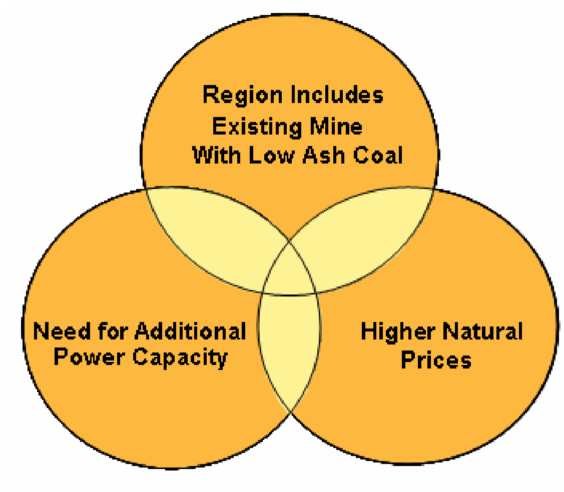


Figure 54. Scenario for Development of Long Range Target Market

Stage 1: Scale Up to First Commercial Installation

A prospective coal slurry supplier such as CQ Inc. will start with a naturally occurring low-ash coal (so as to minimize coal cleaning costs), and secure finances to build and operate a 100,000 to 200,000 ton/year slurry plant. This plant will be located at an existing operating mine so as to take advantage of pre-existing coal prep, storage, and washing facilities. The timing for building this first slurry plant will be governed by natural gas prices in the U.S. and/or heavy fuel oil prices internationally, whichever rises to the favorable range (for CWF) first. The plant economics are projected at $1.50 per million Btu process cost, including labor, capital, electricity, additives, O&M. Depending on the price of coal feedstock, the clean coal fuel slurry is projected to cost $3.50 to $3.70 per million BTU, delivered to the coal diesel site. The initial coal slurry process plant will be intentionally sized to operate at thin margins in order to launch the technology. Higher returns are then expected with larger CWF plants in Stage 2. This initial 100-200K ton/year plant will produce only enough CWF to supply 2 to 4 regional coal diesel plants at 25 MW each.

In parallel with the efforts of the coal slurry supplier (probably working under a collaborative marketing arrangement), a major large engine manufacturer such as Fairbanks Morse, GE, or Caterpillar will aggressively and proactively find cogeneration and/or independent power coal diesel gen-set orders for these 2 to 4 power plants. Population centers with relatively high natural gas prices and high demand for electric power capacity will be favored as Fairbanks Morse identifies the target city or cities for the first commercial clean coal diesels to be installed. The ideal city or region for Stage 1 will offer all three attributes:



Stage 1 of the commercialization path might occur as early as year 2015 and as late as year 2030 or beyond, depending on natural gas prices, the demand for on-site 10-100MW power, and other factors. This will be the first market penetration of the coal diesel in the field.

Stage 2: Replication in the U.S. and Internationally

Following Stage 1, the Clean Coal Diesel technology and engine-grade CWF process plant will be replicated, generally at larger sizes. We call this Stage 2 of the commercialization path. Each replication will consist of one new CWF process plant serving several new coal diesel power plants located in the same geographic region. Whereas the initial coal slurry supplier (e.g. CQ Inc.) would build and operate the Stage 1 plant, for these larger plants in Stage 2, the initial slurry supplier would license the technology to major oil and/or coal companies in the U.S. and internationally. Typically the CWF process plants will be sized at 1 million tons per year (5X to 10X greater than that of Stage 1) in order to optimize costs and infrastructure. Each such CWF plant will support twenty (20) regional coal diesel plants at 25 MW each, located within about an 80 to 150 mile radius of the CWF plant to allow tank truck delivery of CWF.

Depending on the dynamics of the energy markets, the replication of coal diesel plants might proceed more rapidly overseas, where stationary diesels commonly utilize heavy fuel oil (while in the U.S., gas is burned). For example, very low ash coal suitable for engine grade

CWF is prevalent in Indonesia and New Zealand. The coal slurry supplier might elect to contract with BHP, for example (a major player in the Australian mining business), to build one million ton per year CWF process plants.

In parallel, the coal engine manufacturer would gear up to meet the initial market demand of 10 engines per year, then 30 engines per year, and even 100 engines per year. Again, a cooperative marketing agreement with the specialty fuel supplier would facilitate the evolution of the market. However, at some point, the initial US based engine manufacturer might elect to license the technology to MAN or Wartsila, the two major primary-based engine conglomerate building stationary and marine engines. MAN and Wartsila (and their licenses) control over 80% of the stationary engine market worldwide. For example, Fairbanks Morse (the manufacturer who participated in this demonstration) has excellent contacts with MAN and in fact is a licensee of MAN.

The ultimate potential for the coal diesel is as follows:

- **60 CWF Plants U.S. and worldwide**
- **60 million tons/year (1% of world coal consumption)**
- **30,000 MW capacity**
- **150 billion kWh (assuming 5,000 hours full load equivalent)**
- **0.8 - 1.2% of worldwide generation**

6.2 Commercialization Barriers and Competitive Technology

The two-stage scenario described above brings us to the major commercialization barriers and how the coal supplier and engine manufacturer would plan to overcome them. Additional commercialization issues that need to be addressed include:

a) Competitive technology, and
b) Feasibility of retrofitting existing stationary diesel-engine power plants.
c) Fuel cost breakeven point for CWF vs. oil or natural gas.

Competitive Technologies for 10-100 MW On-Site Power
Other on-site power technologies in the 10-100MW range that the coal-diesel will compete against (along with their own future challenges) include:

- Natural gas reciprocating engine generator sets (one future challenge is prohibitive natural gas price and short supply)
- Gas turbines (natural gas) which are particularly competitive above 30MW plant size (future challenge—lower efficiency than reciprocating engines at partial load, plus prohibitive natural gas prices).
- Smaller clean coal boiler systems based on fluidized bed (future challenge—capital cost is higher than coal diesel, and on-site coal storage is a disadvantage).
- Reciprocating and Gas Turbine generator sets operated on coal derived synthetic fuel (future challenge—synthetic fuel from coal likely to cost more [delivered to site] than engine grade CWF).

A complete report summarizing the competitive technologies to the coal diesel was published in 1992 (see ref. 91). In summary, the coal diesel is projected to be very competitive against all of these technologies.

Feasibility of Retrofitting Existing Fleet of Stationary engines

Regarding retrofit of existing stationary engines to CWF, this has been examined in detail and shown to be feasible only on a limited basis. The reasons are that:

- Most stationary recips in the U.S. are gas fueled, and these engines are not set up with liquid fuel storage tanks and injection equipment. Also many natural gas engines have lower compression ratio than diesels and therefore are not designed for the cylinder pressures of coal diesel operation.
- Only heavy-fueled engines are suitable for CWF retrofit. The cost of the retrofit including engine parts, emission controls, and CWF tankage would be 50-60% of the complete new coal diesel engine system.

Fuel Cost Breakeven Point for CWF vs. Fuel Oil or Natural Gas

The question of fuel cost breakeven point for CWF can be addressed by (a) first establishing a “baseline” typical comparison between operating a given 10 MW diesel power plant on CWF vs. conventional fuel, then (b) studying the impact of changing the key assumptions such as CWF transportation distance, source coal cost, power plant load factor and price of conventional fuel. Based on the analysis and estimates presented above in Section 5, the price of CWF is assumed to be $3.50 per million BTU including nominal delivery cost.

6.3 Patentable Components and Suggested Manufacturers

The primary items that are patentable with the CWF injection system would be the explicit know how as to the materials to be used, the tolerances required, type of hard coatings and thickness. The following components and their areas on know how

Item 1) Shuttle body

This component has particular areas of the design which requires certain know how in order to properly design the assembly. The shuttle to bore clearance specific to the application and design according to the CWF particle size used, the shuttle piston upper and lower sealing lands are design to achieved a controlled leakage in order to maintain a proper oil barrier between the CWF and the shuttle to bore clearance. This is critical to assure the CWF does not migrate into the clearance between and during the injection cycle.

Item 2) CWF nozzle assembly

The design has two areas which require particular expertise in the assembly of its components. The first is the assembly of the tungsten carbide (WC) valve seat to the main body. This requires specific know-how as to the hardness level of the main body and the method of securing the seat to the main body.

The second component is the needle valve; this design requires certain expertise as to the assembly of the upper steel valve stop and lower WC valve. A precise technique is required in order that the upper steel component does not fracture the WC item during assembly. Also the type and thickness of the hard coatings used to improve the lubricity and reduction of the wear.

Item 3) CWF check valve assembly

Because of the assembly technique of the valve seat to the main body, this assembly is the same as the nozzle body. Caution is required to reduce the possibility of the seat cracking during the assembly of the seat to the main body and during the normal operation of the check valve assembly.

Candidate firms to be licensed to build the coal injectors

The companies that would be best suited to manufacture the main injector would be OMT SpA, who has manufactured this size injector since 1936 and are current manufacturing injectors for the Pielstick engine. Other manufactures that are possibility are L'Orange, DUAP or Woodward. Based on industrial experience the listing of these particular vendors are as to the level of quality with OMT as the highest.

The pilot injector could possibly be manufactured by the same firms.

The companies listed above would be capable to manufacture the main nozzle body for the complete injector as well as spares. However, the manufacture of the pilot nozzle tip, FIRAD SpA would be capable to produce the small 17mm type nozzle tip design.

7. Outlook

The key practical implications of this project were as follows:

- Test results show that the technology met the efficiency and emissions targets, and performance in these areas did not degrade during the tests. Therefore, efficiency and emissions improvements areas are not on the critical path; straightforward engineering effort can achieve scale-up of the engine and emission system to commercial plant sizes.

- Longer run times are needed to estimate useful lifetimes of certain engine components, particularly the useful life of piston rings and exhaust valves. This data on engine components is critical before commercial introduction of the technology. Engineering solutions and material selections are available for durable components, but these solutions must be optimized and demonstrated for several thousand hours.

Thus, the next step toward commercialization is a field demonstration program with 6,000 hours of engine run time on coal fuel. Since this will require three years (the program will include several lengthy test periods, rather than continuous operation), the implication is that commercial introduction (plant orders) can be targeted in the post-2010 timeframe, assuming a successful field demonstration and a favorable fuel price structure.

Coal slurry fuel is expected to become competitive in the U.S. with diesel oil and natural gas in the 2010-2025 timeframe, based on energy price projections made by DOE and others. This gives manufacturers the necessary time to optimize and demonstrate the wear solutions for critical hard parts, through a field demonstration program.

Additional field demonstration opportunities for small coal-diesel plants will be pursued in special situations where clean coal slurry holds a price advantage, such as:

- Alaska rural electrification (where diesel oil costs $4-$12 per million Btu delivered to certain remote communities).
- China, which has both coal reserves and the need for rapid installation of non-grid power (such as diesels).
- Eastern Europe, which also has coal reserves and is undergoing rebuilding of the electric power infrastructure so as to greatly reduce emissions.

Test experience has shown that the capital cost of the coal diesel plant will not be a barrier to commercialization. The cost of all equipment modules for the plant has been established, and the installed plant cost estimates appear to be competitive:

- $1600/kW for early demonstration plants
- $1300/kW for mature plants
- These costs are well below the capital cost of other small coal plants, especially in the NUG market under 50 MW plant size.

Test results have established the coal-water slurry specification, and have proved that a wide range of coals can be utilized to prepare engine-grade slurry. The cost of the slurry will be under $2.00/MMBtu plus raw coal feedstock cost once adequate slurry-demand exists in a given region. The commercialization plan incorporates a series of steps to build up an "infrastructure" for coal-water slurry production and distribution. This is recognized as critical.

The Clean Coal Diesel Plant of the future is targeted for the 10-100 MW non-utility generation (NUG) and small utility markets, including independent power producers (IPP) and cogeneration. A family of modular plant designs will be offered. While a plant can be built as small as 2 MW, our cost projections indicate that an 8 MW plant is likely to be at the lower end of what is economically attractive. It is quite realistic to design and build clean coal diesel plants in the 50-150 MW capacity range. In fact, the 14-25 MW class diesel engines that would be utilized in these larger plants offer a fuel savings advantage over the smaller engines (typically 45% vs. 40% simple cycle efficiency, LHV). It should also be noted that the coal diesel plant also can be configured for cogeneration applications.

The Clean Coal Diesel will offer the following performance characteristics in its mature configuration beginning in the 2015-2030 timeframe:

- Installed cost $1300/kW
- Efficiency 48.2% (LHV)
- NO_x emissions 0.20 lb/MMBtu
- SO_x emissions 0.08 lb/MMBtu
- Particulate emissions 0.003 lb/MMBtu

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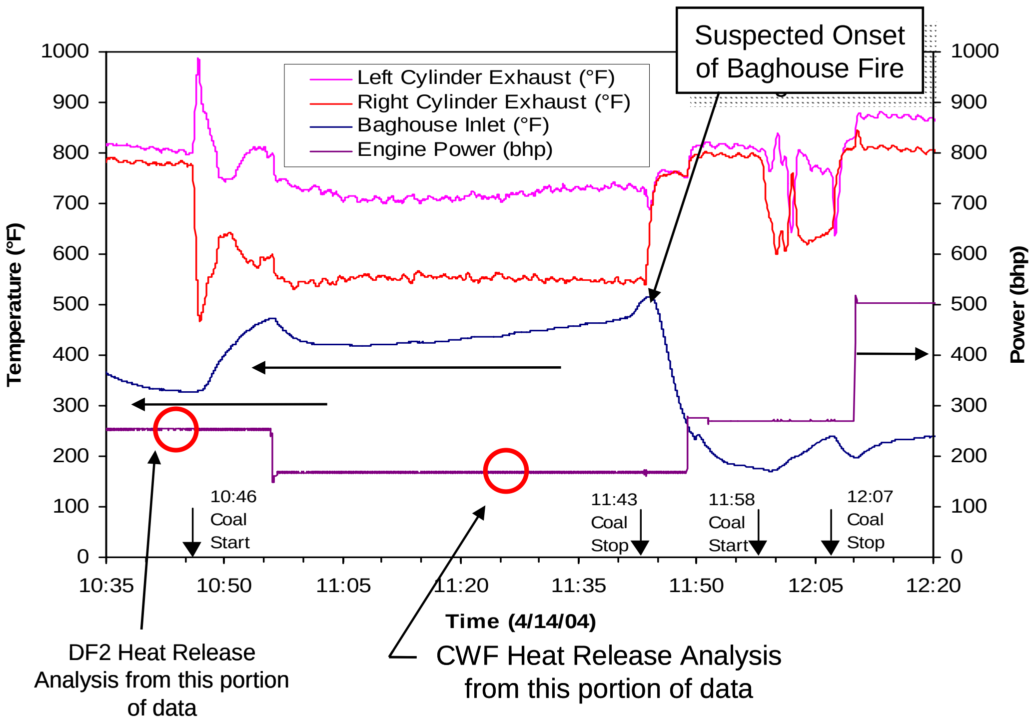
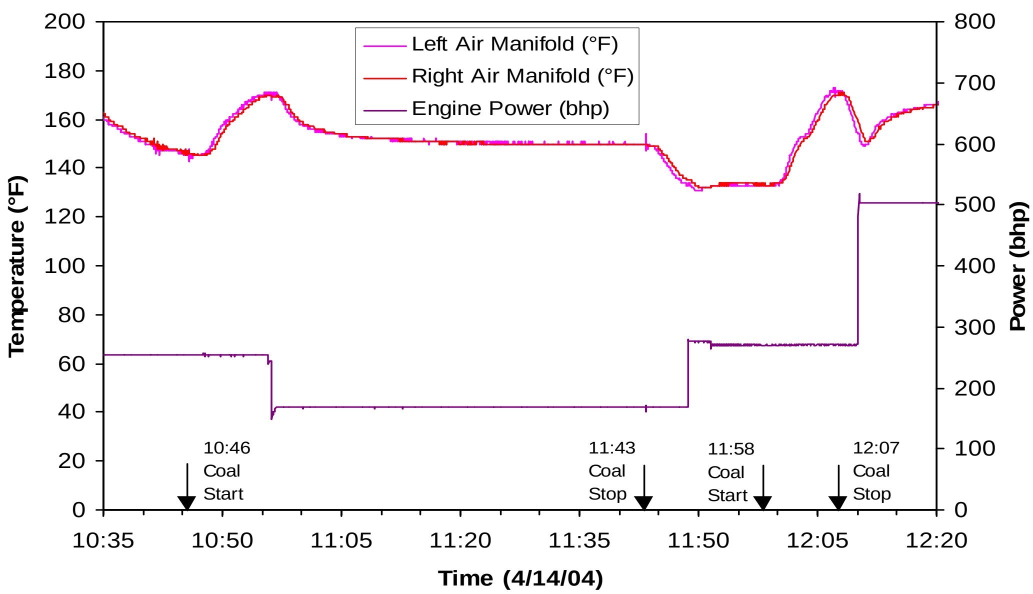
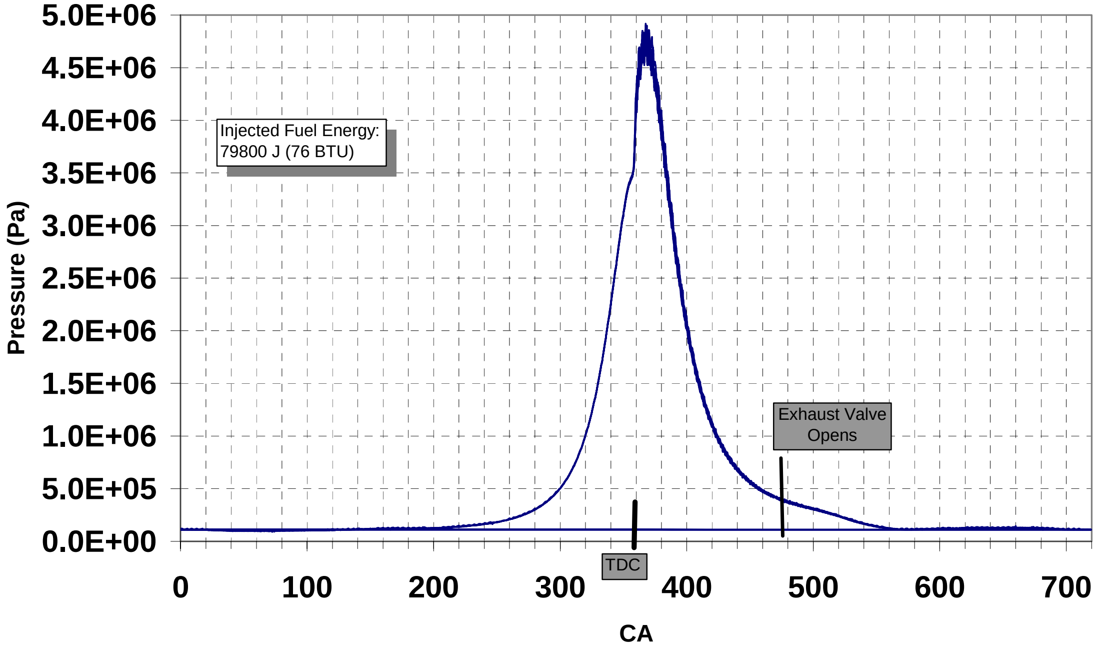
The time history of the successful run shows a difference in exhaust temperatures, which can be attributed to incomplete combustion in one cylinder



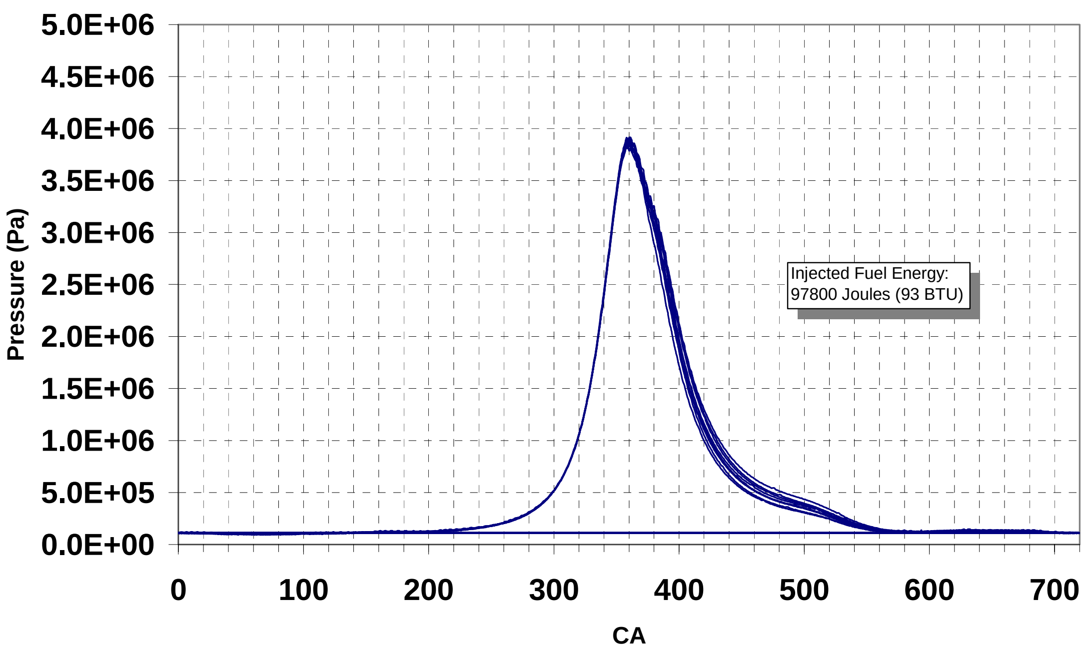


Comparing the DF2 and CWF traces for the left cylinder of the engine highlights the need for pilot timing adjustment

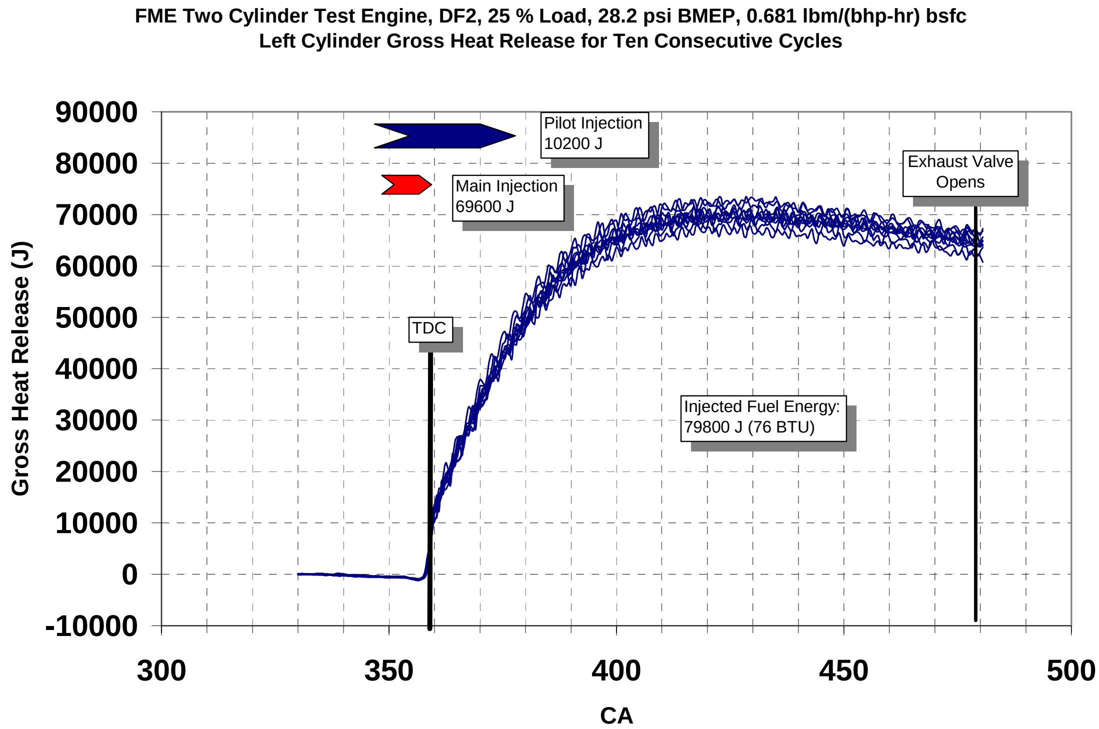
FME Two Cylinder Test Engine, DF2, 25 % Load, 28.2 psi BMEP, 0.681 lbm/(bhp-hr) bsfc
Left Cylinder Pressure Trace for Ten Consecutive Cycles



FME 2 Cylinder Test Engine, Usibelli CWF, 17% Load, 18.7 psi BMEP, 1.75 lbm/(bhp-hr) bsfc
Left Cylinder Pressure Trace for Ten Consecutive Cycles

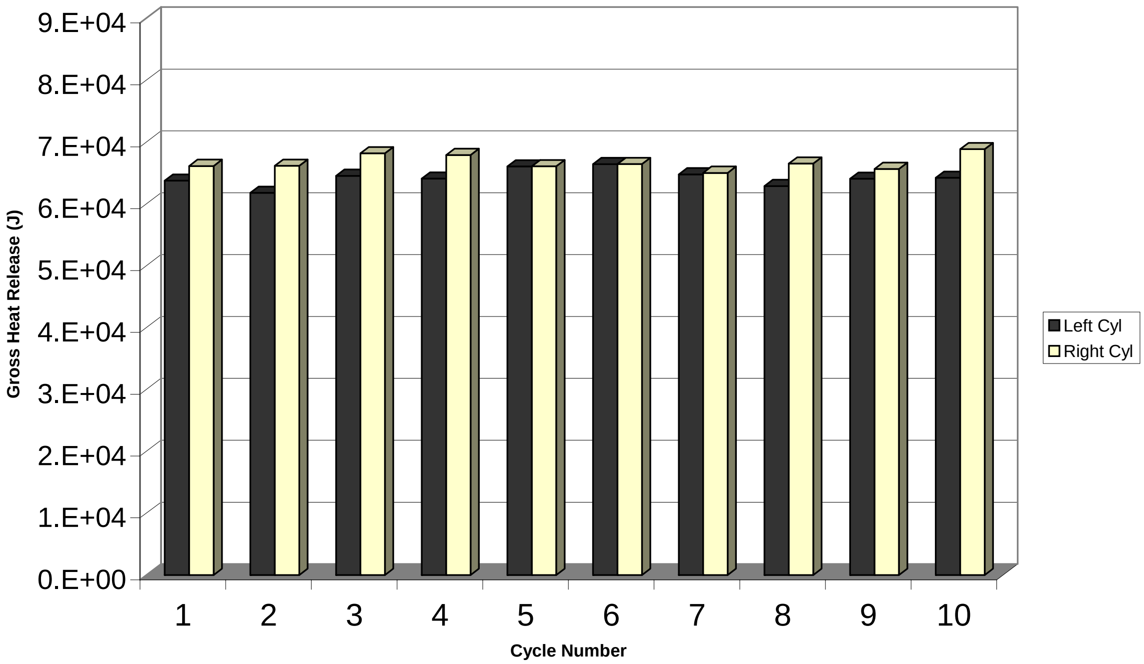


DF2 Heat Release shows repeatable combustion, whereas the CWF heat release shows incomplete in-cylinder combustion

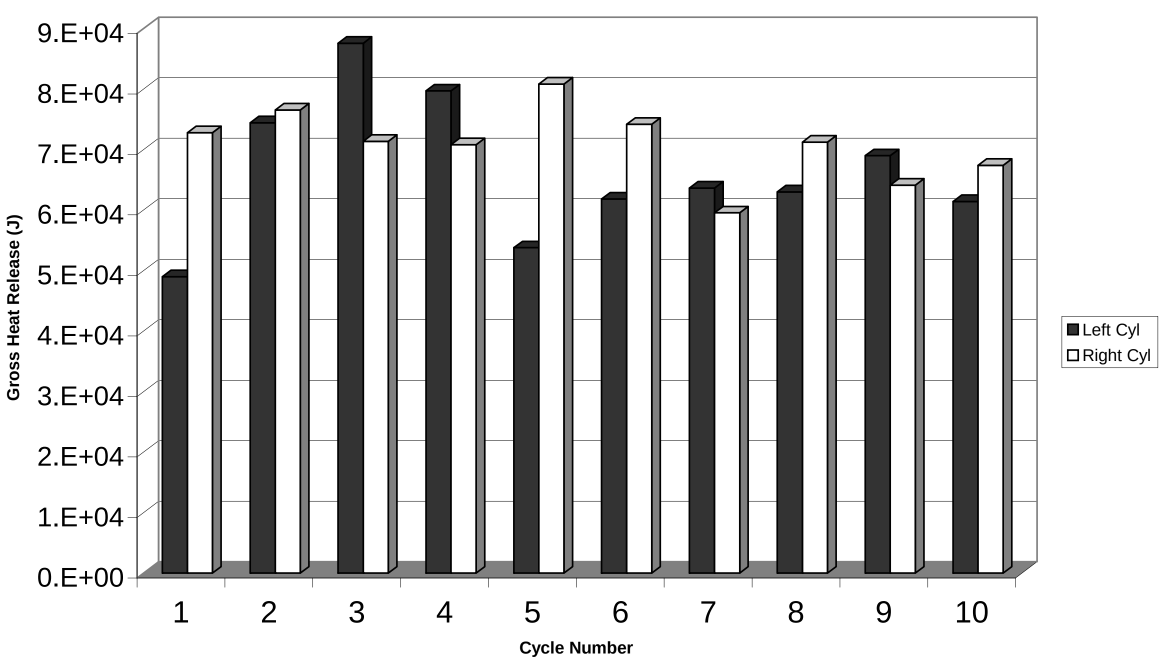


DF2 Heat Release shows repeatable combustion, whereas the CWF heat release shows erratic in-cylinder combustion

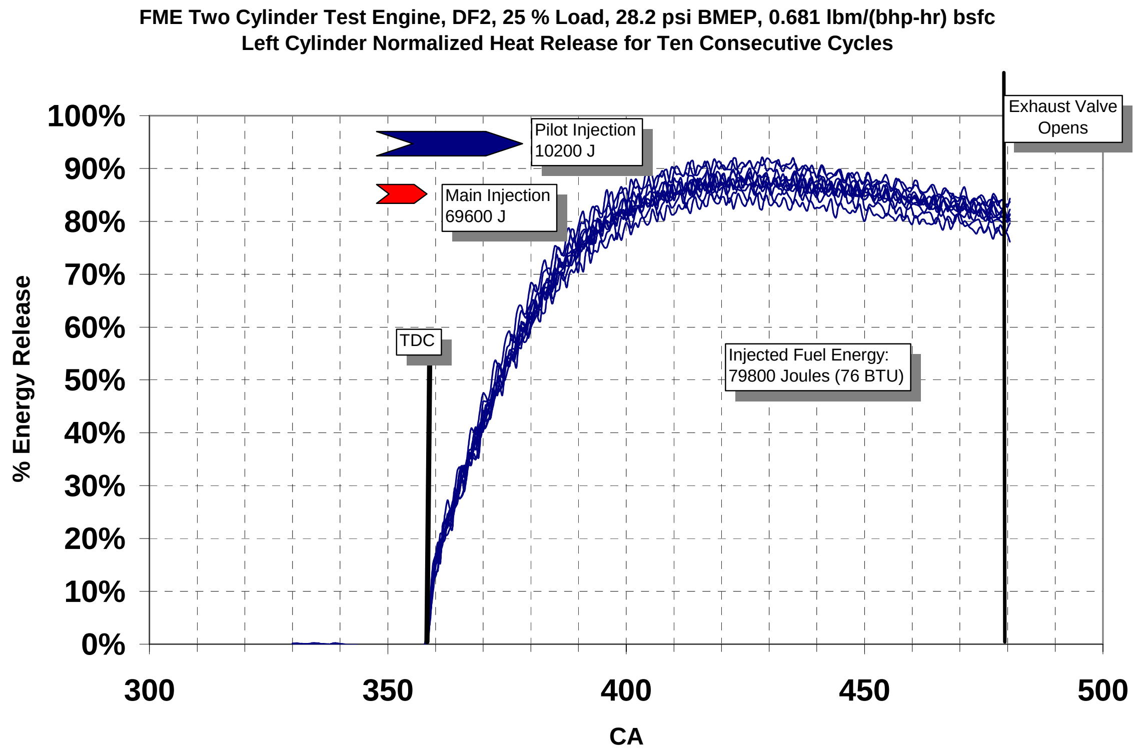
FME Two Cylinder Test Engine, DF2, 25 % Load, 28.2 psi BMEP, 0.681 lbm/(bhp-hr) bsfc
Gross Heat Release Comparison for Ten Consecutive Cycles

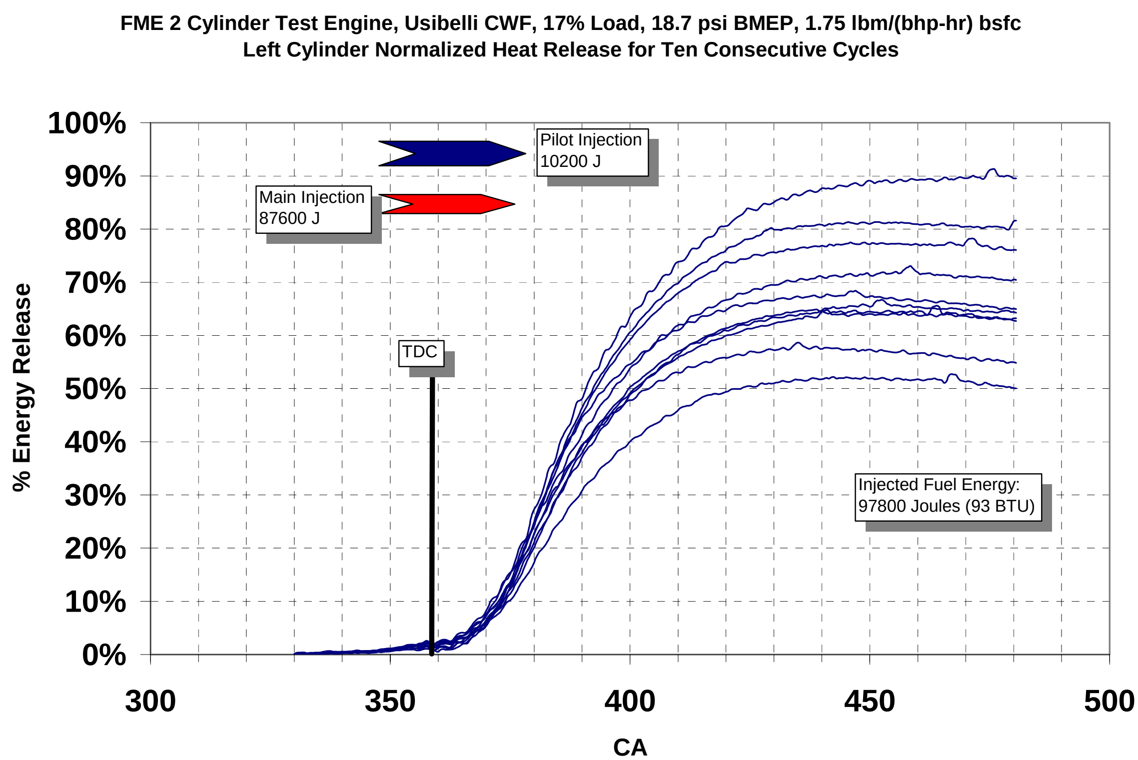


FME 2 Cylinder Test Engine, Usibelli CWF, 17% Load, 18.7 psi BMEP, 1.75 lbm/(bhp-hr) bsfc
Gross Heat Release Comparison for Ten Consecutive Cycles



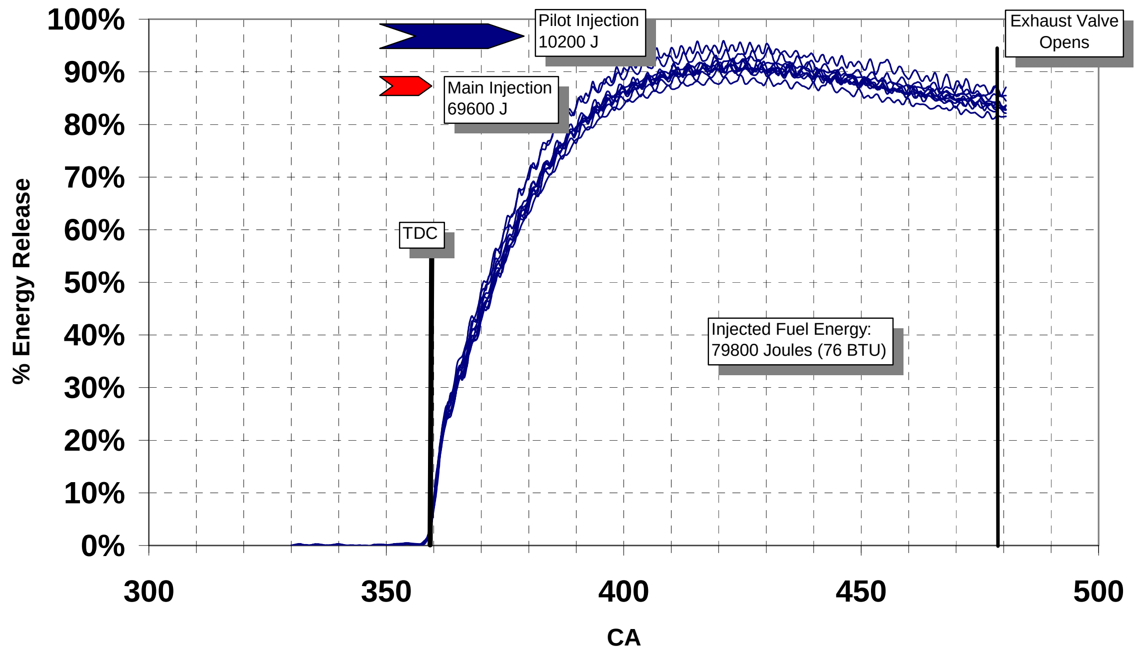
Examining the energy release as a fraction of injected fuel energy shows that the DF2 burns close to 90%, but the CWF fuel burns 50-90%



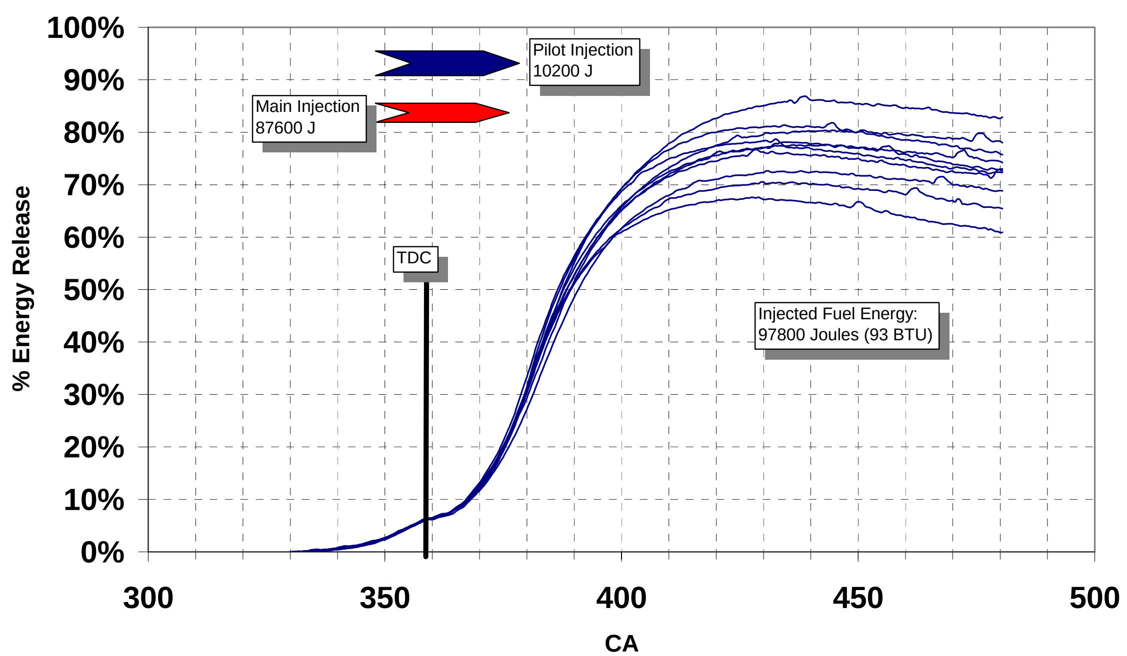


The right cylinder CWF energy release again shows better repeatability (68-85%) in energy release, but still not as repeatable as DF2

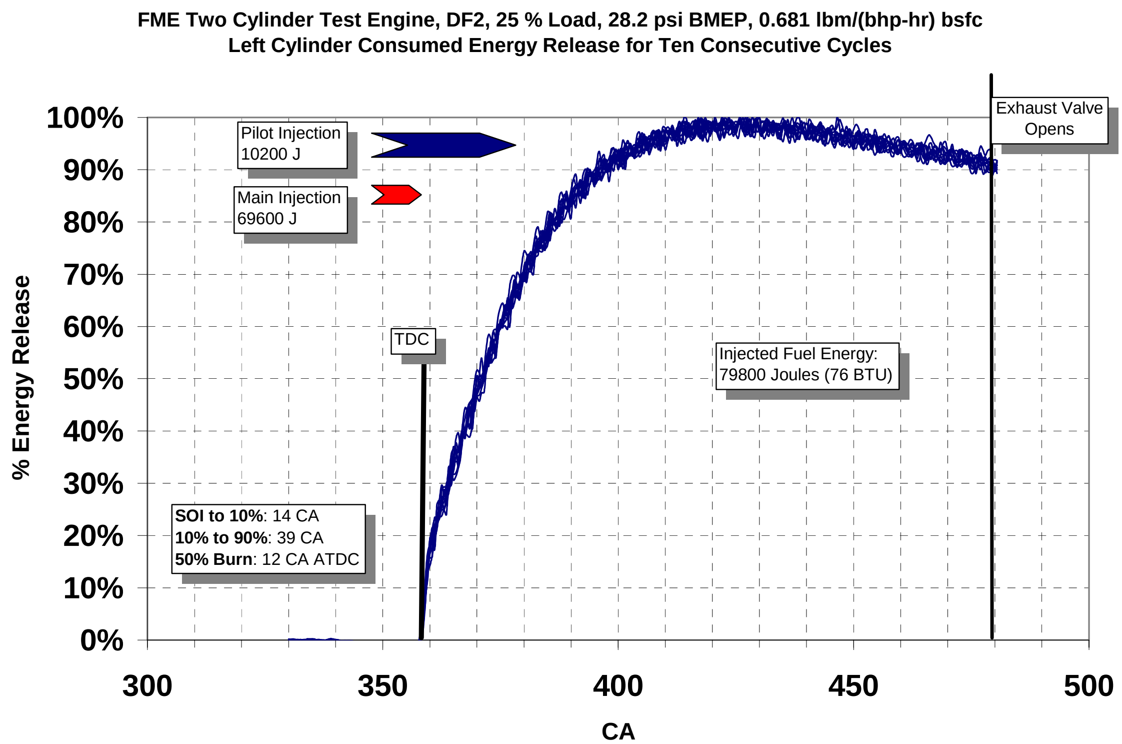
FME Two Cylinder Test Engine, DF2, 25 % Load, 28.2 psi BMEP, 0.681 lbm/(bhp-hr) bsfc Right Cylinder Normalized Heat Release for Ten Consecutive Cycles

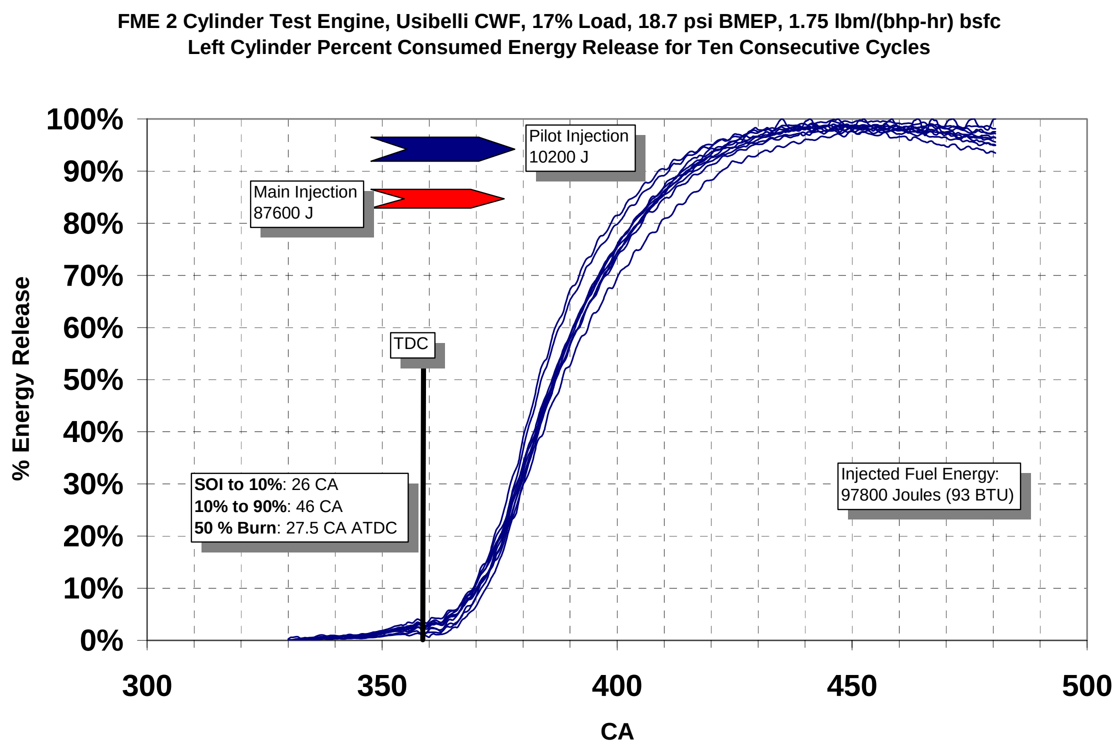


FME 2 Cylinder Test Engine, Usibelli CWF, 17% Load, 18.7 psi BMEP, 1.75 lbm/(bhp-hr) bsfc Right Cylinder Normalized Heat Release for Ten Consecutive Cycles



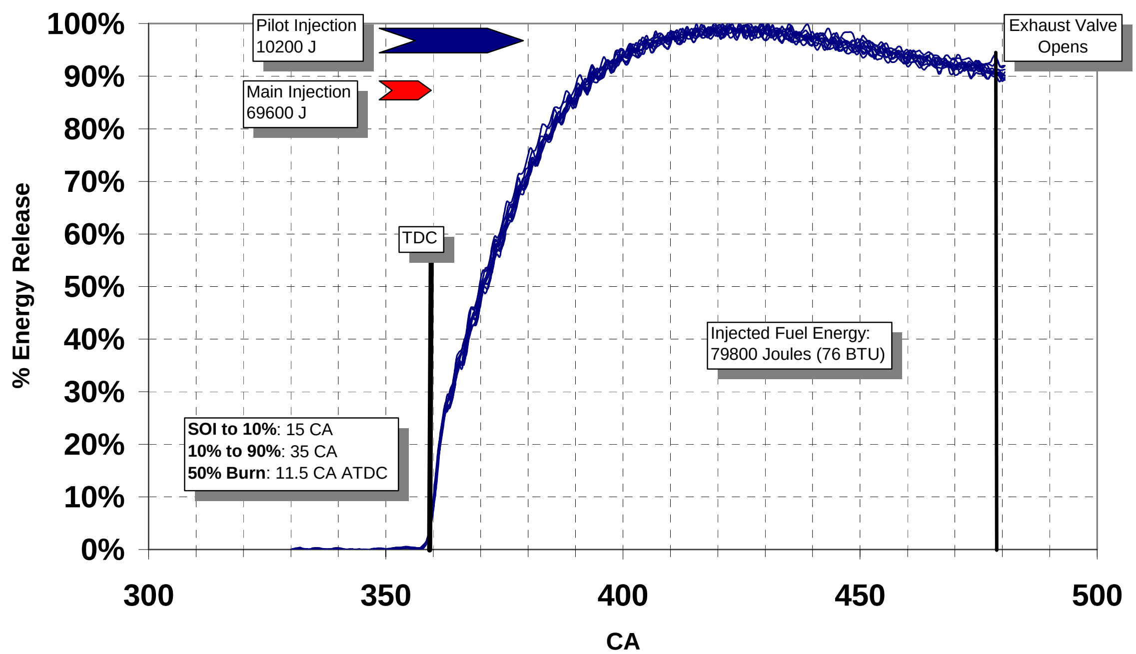
Comparing the percent of fuel injected that was actually burned, it can be seen the burn rates of DF2 are fairly repeatable, but CWF takes longer to ignite and slightly longer to burn



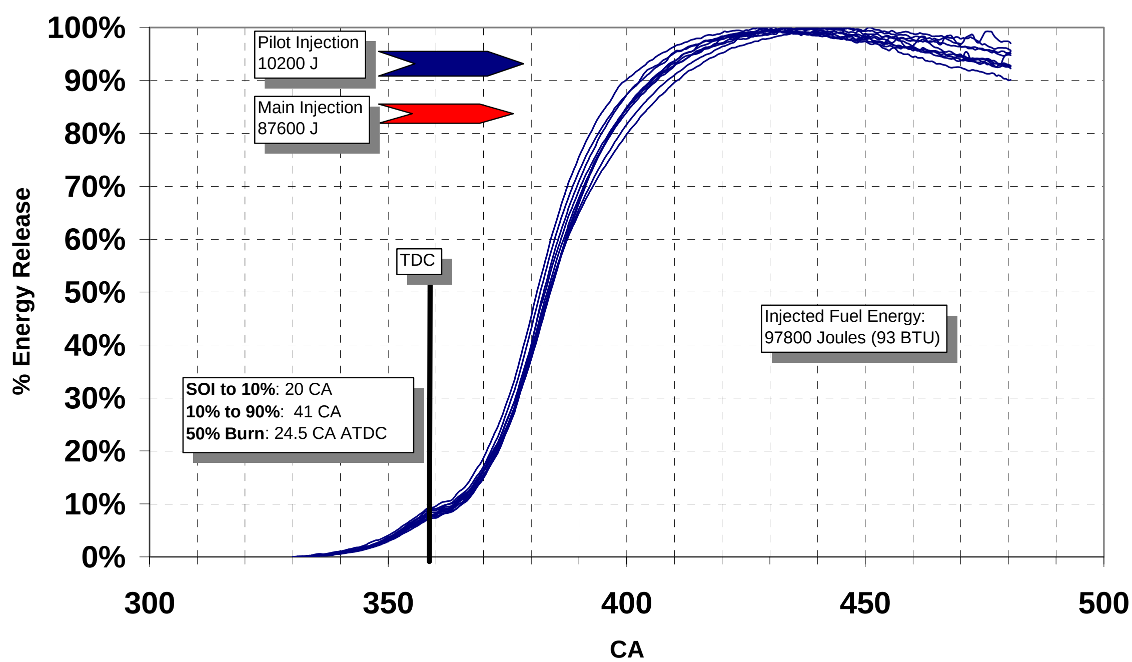


The right cylinder shows similar results—the CWF ignites slower and takes slightly longer to burn

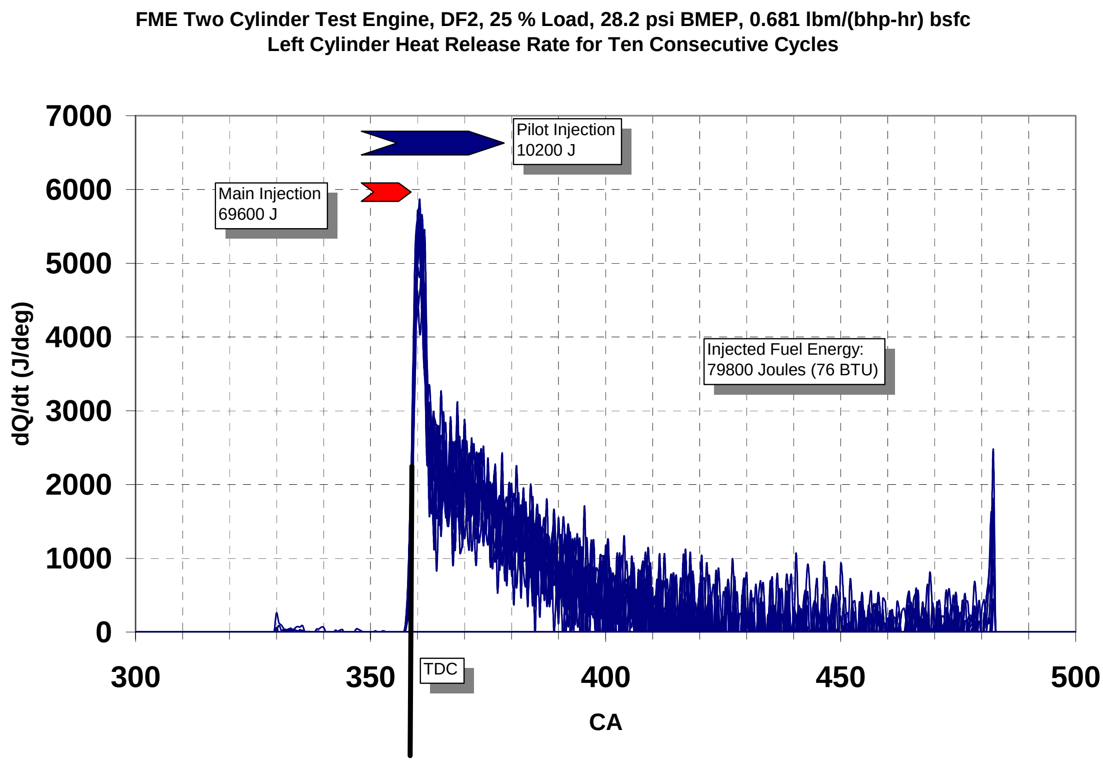
FME Two Cylinder Test Engine, DF2, 25 % Load, 28.2 psi BMEP, 0.681 lbm/(bhp-hr) bsfc Right Cylinder Percent Consumed Energy Release for Ten Consecutive Cycles

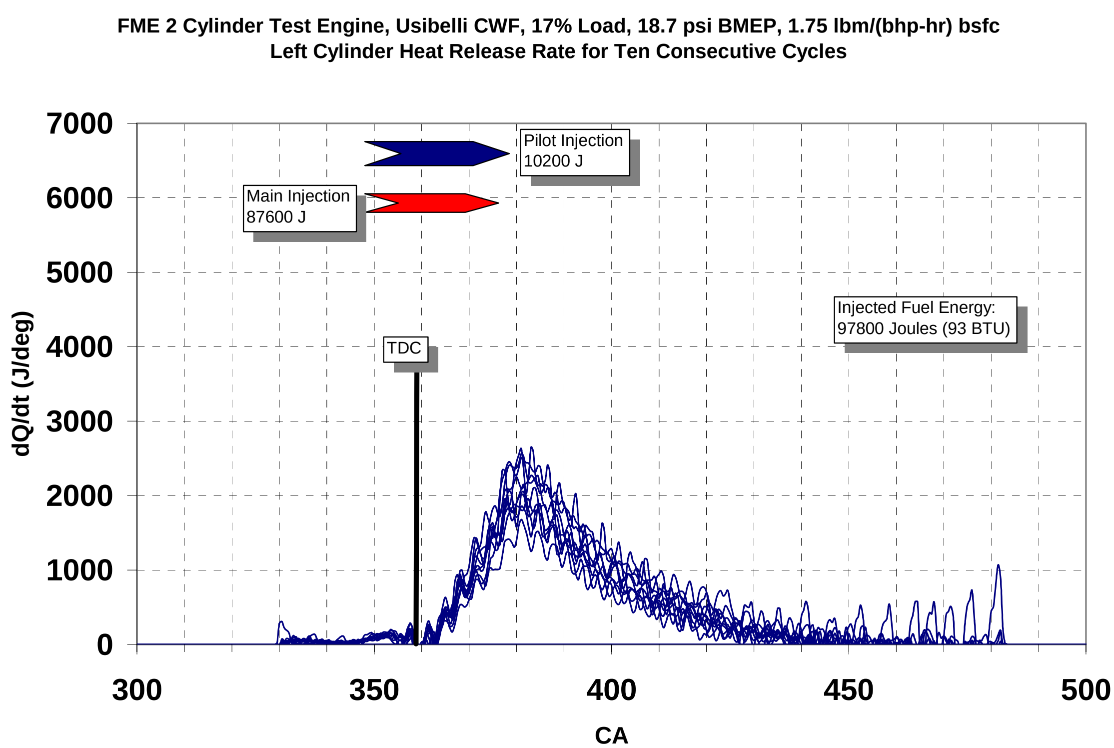


FME 2 Cylinder Test Engine, Usibelli CWF, 17% Load, 18.7 psi BMEP, 1.75 lbm/(bhp-hr) bsfc Right Cylinder Percent Consumed Energy Release for Ten Consecutive Cycles



The rate of heat release reveals the problem with the late pilot injection—DF2 combustion shows pre-mixed combustion initially and then a diffusion flame, whereas the CWF is 100% mixing controlled combustion

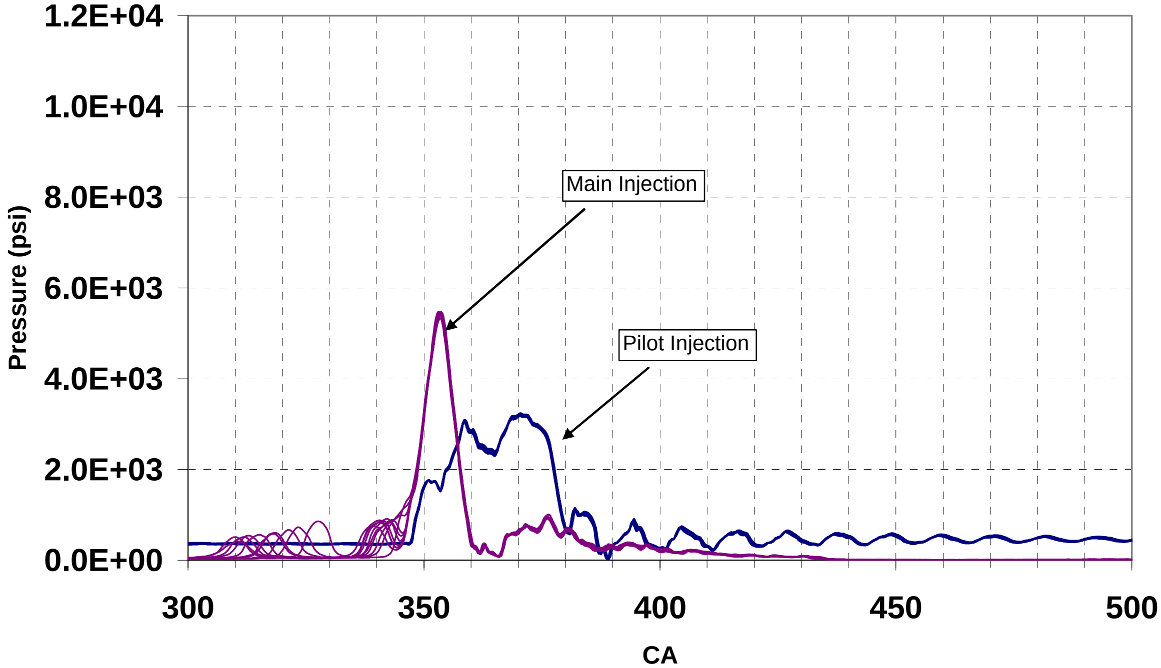




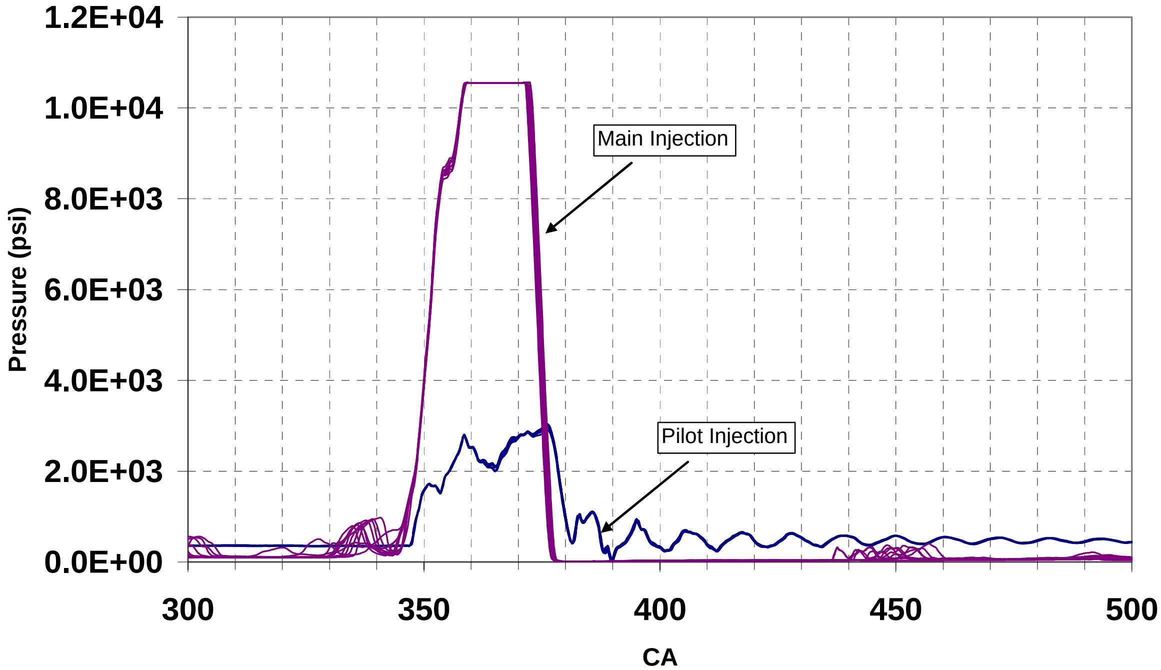
The right cylinder shows similar results—the DF2 initial heat release spike is pre-mixed combustion, compared to the CWF, which is 100% mixing controlled

The injector pressure traces show excellent, repeatable pressure profiles

FME Two Cylinder Test Engine, DF2, 25 % Load, 28.2 psi BMEP, 0.681 lbm/(bhp-hr) bsfc
Left Cylinder Injection Pressure Trace for Ten Consecutive Cycles

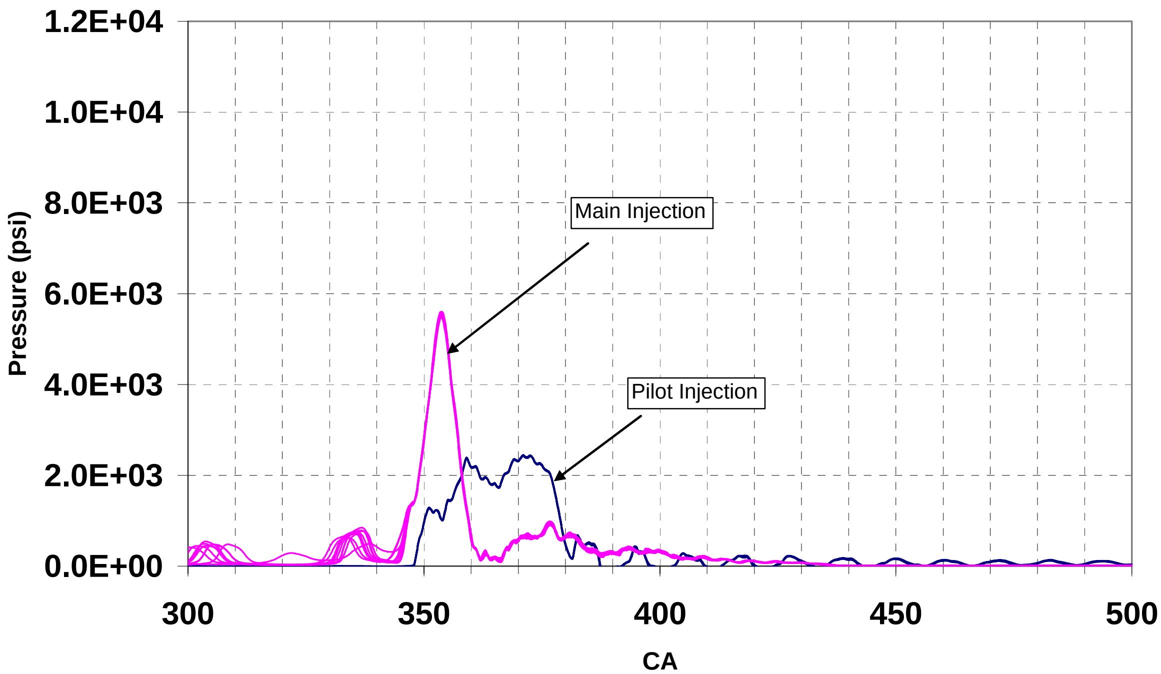


FME 2 Cylinder Test Engine, Usibelli CWF, 17% Load, 18.7 psi BMEP, 1.75 lbm/(bhp-hr) bsfc
Left Cylinder Injection Pressure Trace for Ten Consecutive Cycles

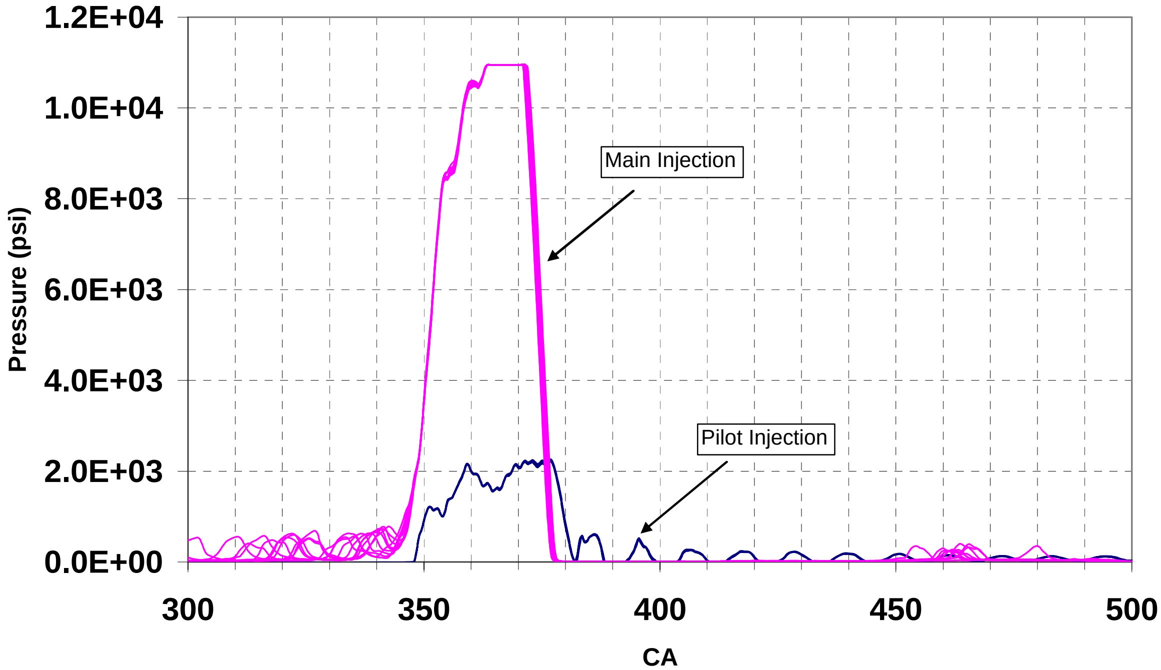


The right cylinder injection profile shows similar repeatability to the left cylinder

FME Two Cylinder Test Engine, DF2, 25 % Load, 28.2 psi BMEP, 0.681 lbm/(bhp-hr) bsfc
Right Cylinder Injection Pressure Trace for Ten Consecutive Cycles

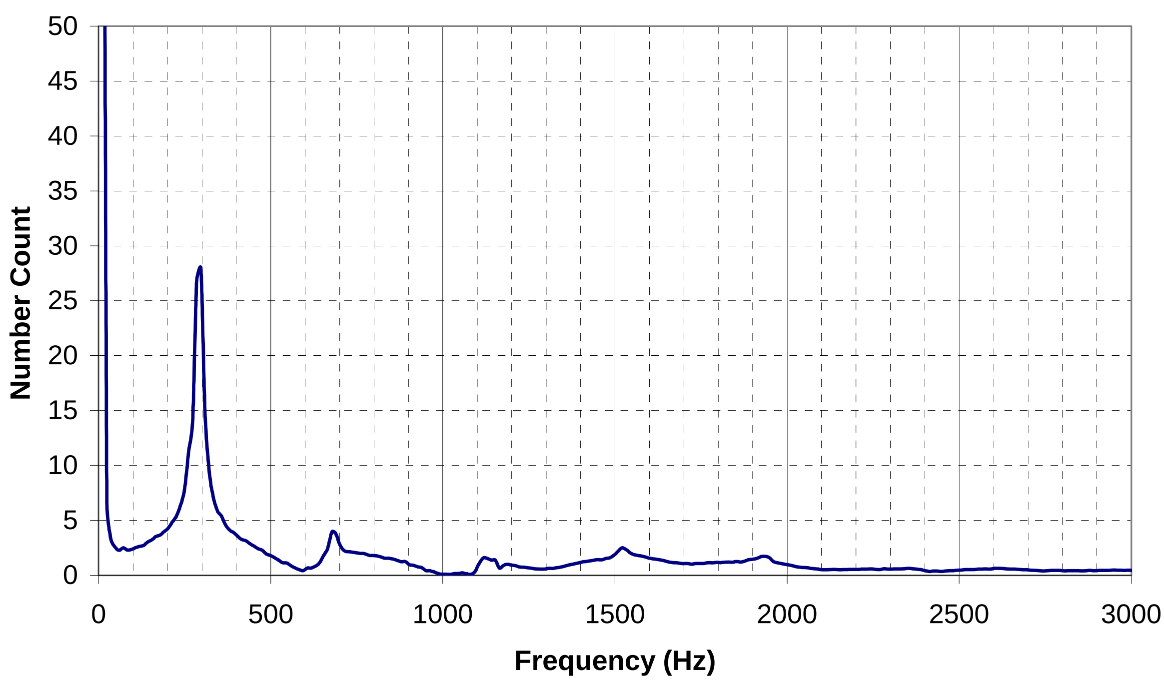


FME 2 Cylinder Test Engine, Usibelli CWF, 17% Load, 18.7 psi BMEP, 1.75 lbm/(bhp-hr) bsfc
Right Cylinder Injection Pressure Trace for Ten Consecutive Cycles

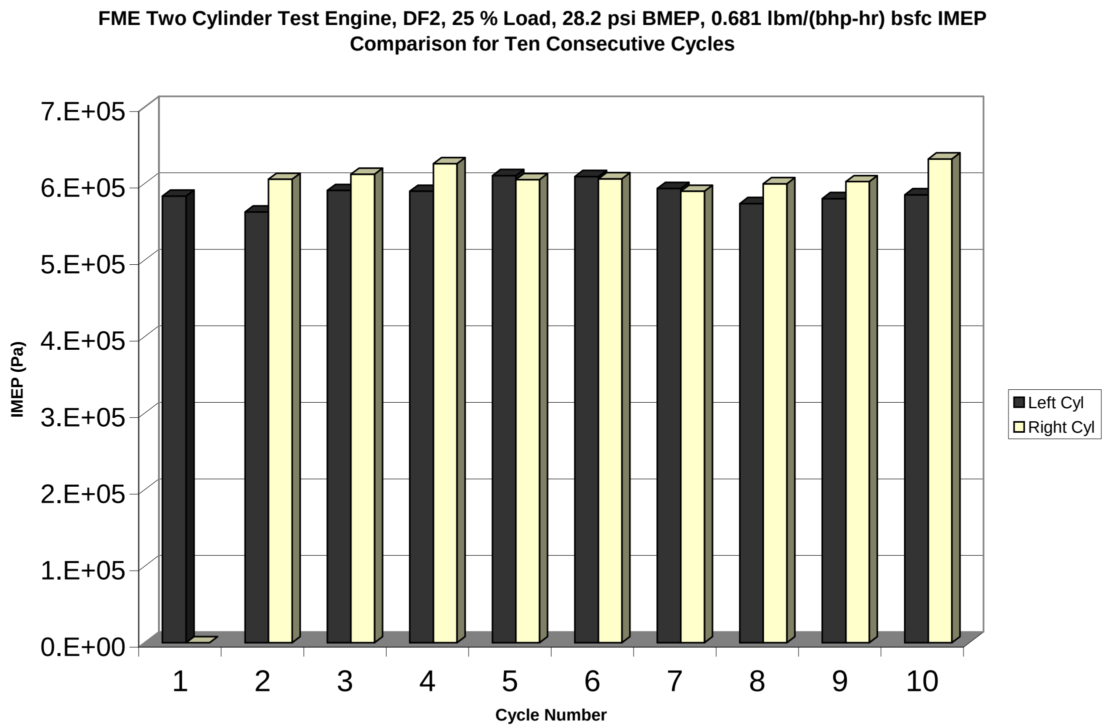


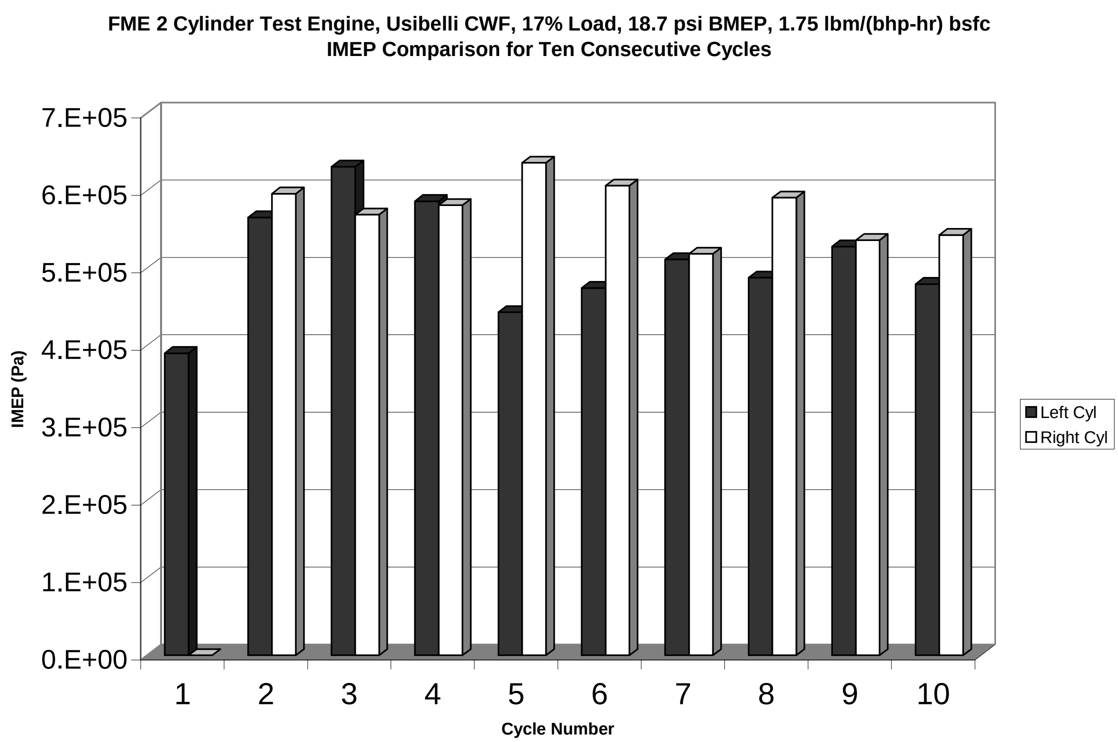
Fourier Analysis of the pressure wave seen after the pilot injection occurs shows a frequency of 297 Hz, which is well above engine operating frequency

Fourier Transform of Pressure Wave Seen In Pilot Injector Signal

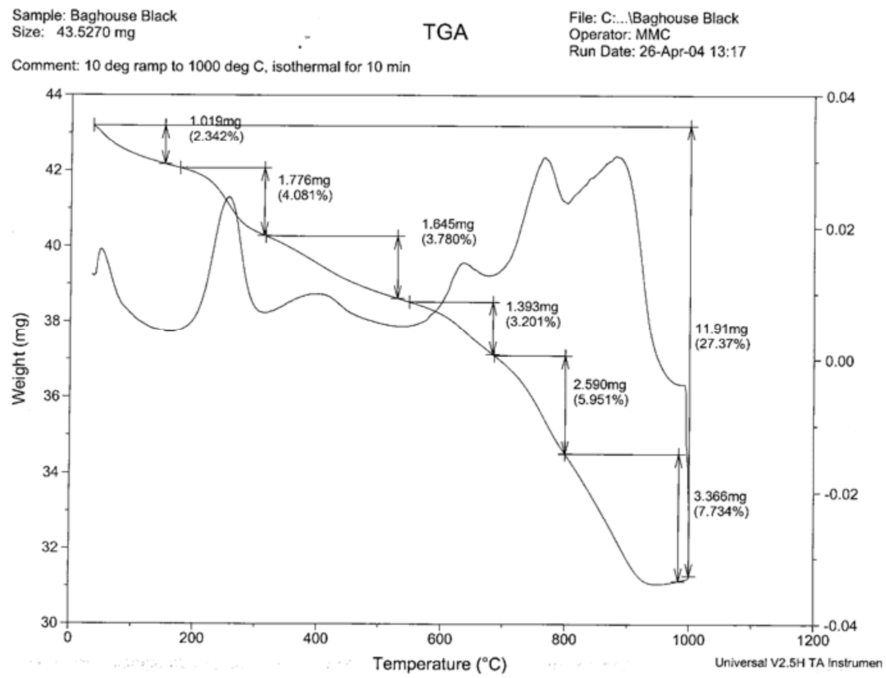


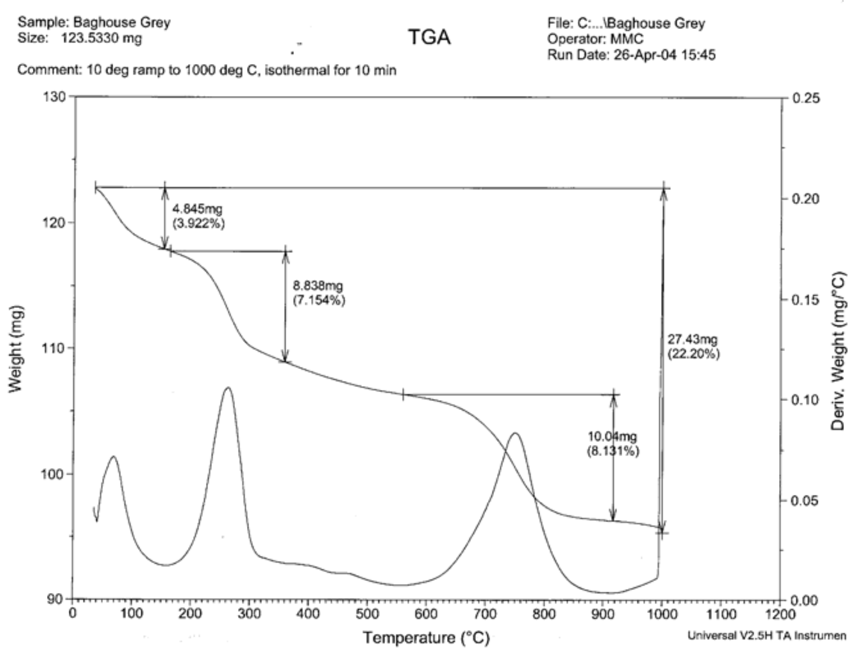
Looking at the repeatability of Indicated Mean Effective Pressure (IMEP), DF2 shows better repeatability, but this is assumed due to the timing of the pilot injection for CWF





TGA Analysis from the solids found in the baghouse show that carbon-containing ash was present, evidenced by the peaks in the derivative of the weight loss

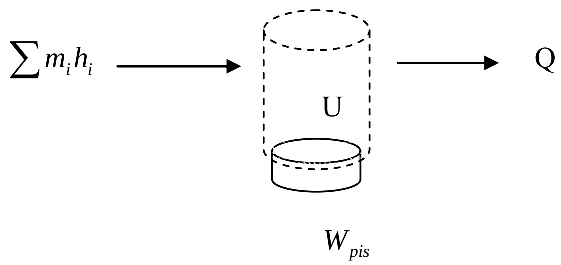




Appendix B. Combustion Analysis of Engine Tests

Heat Release Analysis and Exhaust Mass Flow Documentation

This analysis is based upon a one-zone model of the combustion chamber, valid after the intake valve closes and before the exhaust valve opens. Heat release analysis of the in-cylinder pressure signal is based upon the first law of thermodynamics for an open system based upon the following figure:



Assuming a quasi static open system, the first law of thermodynamics may be applied:

$$\frac{dQ_{ch}}{dt} + \frac{dQ_{ht}}{dt} - p\frac{dV}{dt} + \sum m_i h_i = \frac{dU}{dt} \quad (1)$$

The net heat release rate only includes the work done on the piston and the change in internal energy of the cylinder and neglects heat transfer. Since the enthalpy of the fuel entering the cylinder is negligible, and assuming ideal gas behavior, equation 1 becomes:

$$\frac{dQ_{ch}}{dt} = p\frac{dV}{dt} + mc_v\frac{dT}{dt} \quad (2)$$

Since the ultimate goal of the heat release analysis is to derive the heat released from the measured pressure signal, the temperature term in equation 2 may be substituted by applying the chain rule to the ideal gas law (pV=mRT), holding R constant:

$$m\frac{dT}{dt} = \frac{1}{R}\left(p\frac{dV}{dt} + V\frac{dP}{dt}\right) \quad (3)$$

Substituting equation 3 into equation 2 and utilizing the relationship that $c_p/c_v=\gamma$, the net heat release now becomes:

$$\frac{dQ_{net}}{dt} = \frac{\gamma}{\gamma-1}p\frac{dV}{dt} + \frac{1}{\gamma-1}V\frac{dp}{dt} \quad (4)$$

γ is defined as [3]:

Equivalence Ratio ϕ	Compression	Combustion	Expansion
0.4	1.415-1.186E-04T	1.312	1.343-3.653E-05T
0.7	1.404-1.299E-04T	1.29	1.290-1.200E-05T
0.9	1.393-1.302E-04T	1.282	1.289-1.523E-05T
1	1.393-1.300E-04T	1.280	1.282-1.349E-05T
1.1	1.388-1.304E-04T	1.283	1.302-2.213E-05T
1.2	1.384-1.307E-04T	1.291	1.299-1.885E-05T

These values of γ were determined from comparing the more rigorous but more computationally intensive two-zone cylinder model (where burned and unburned gas properties are handled individually) to the one-zone model described above. T is the approximate temperature as calculated from the ideal gas equation.

The net heat released is thus the integral of equation 4:

$$Q = \int \frac{dQ_{net}}{dt} dt \approx m_f Q_{LHV} \tag{5}$$

The difference of the integrated heat release rate and the chemical fuel energy input into the cylinder will approximate the amount of heat transferred to the cylinder walls.

The first law may also be used to calculate the heat-up of the exhaust stream due to the incomplete combustion of CWF. Applying equation 1 to a simple control volume applied to the exhaust pipe (ignoring heat transfer), the following equation may be obtained:

$$\Delta T = \frac{\dot{Q}}{\dot{m} c_p} \tag{6}$$

Where Qdot is the heat release rate of the unburnt fuel, mdot is the mass flow rate of the exhaust, and c_p is the specific heat of the exhaust stream. Plugging in the proper values into equation 6 from the summary of operating conditions contained at the outset of this data package, the temperature rise is found to be on the order of 340 deg Fahrenheit assuming 50% of the injected fuel energy is burned. The time for the heat release of the energy contained in the unburnt CWF is derived from the burn rates found in the heat release analysis.

References for Appendix B: References number 93 – 96 (see reference list).

Appendix C. Previous CWF Engine Test Results from Cooper-Bessemer

The CWF engine demonstrated was a modified four stroke diesel engine based on the Cooper-Bessemer LSB production engine (Figure 55). The demonstration engine was a 20-cylinder version with modified block, camshaft, cylinder heads, pistons, and a unique injection system specifically designed and built for CWS (91). Table 22 lists key engine parameters.

Table 22. Key Parameters of the Demonstration Engine

Model	LSVC-20
Bore	15.5 in
Stroke	22 in
Nominal Speed	400 rpm
No. of Cylinders	20
BMEP (nominal)	208 psi
Cycle	4 stroke
Power Output (nominal)	6200 kW

The novel technologies incorporated into this design to allow utilization of coal-water fuels and achieve target component life are as follows:

- modified "fast rate" fuel injection cam
- larger fuel injection jerk pump
- coal-slurry tolerant fuel injection system
- nozzle tip with sapphire inserts
- ceramic-coated piston rings
- ceramic-coated cylinder liner
- ceramic-coated exhaust valves
- ceramic-coated turbocharger blades
- modified engine block

Engine Test Results (1996 Cooper-Bessemer)

In 1996, the project team completed the initial series of coal-water fuel performance tests on Cooper's full-scale LSC-6 engine at their Mt. Vernon, Ohio test facility (91). This engine was operated with one cylinder burning Ohio CWF and the remaining five firing diesel fuel. About 34,000 pounds of CWF were consumed. Individual cylinder instrumentation allowed us to monitor the performance of the CWF cylinder. In these tests, the fuel efficiency, exhaust temperature and peak cylinder pressure for CWF firing were all within desired ranges. Overall, these test results indicate that Ohio CWF meets our requirements for satisfactory engine performance.

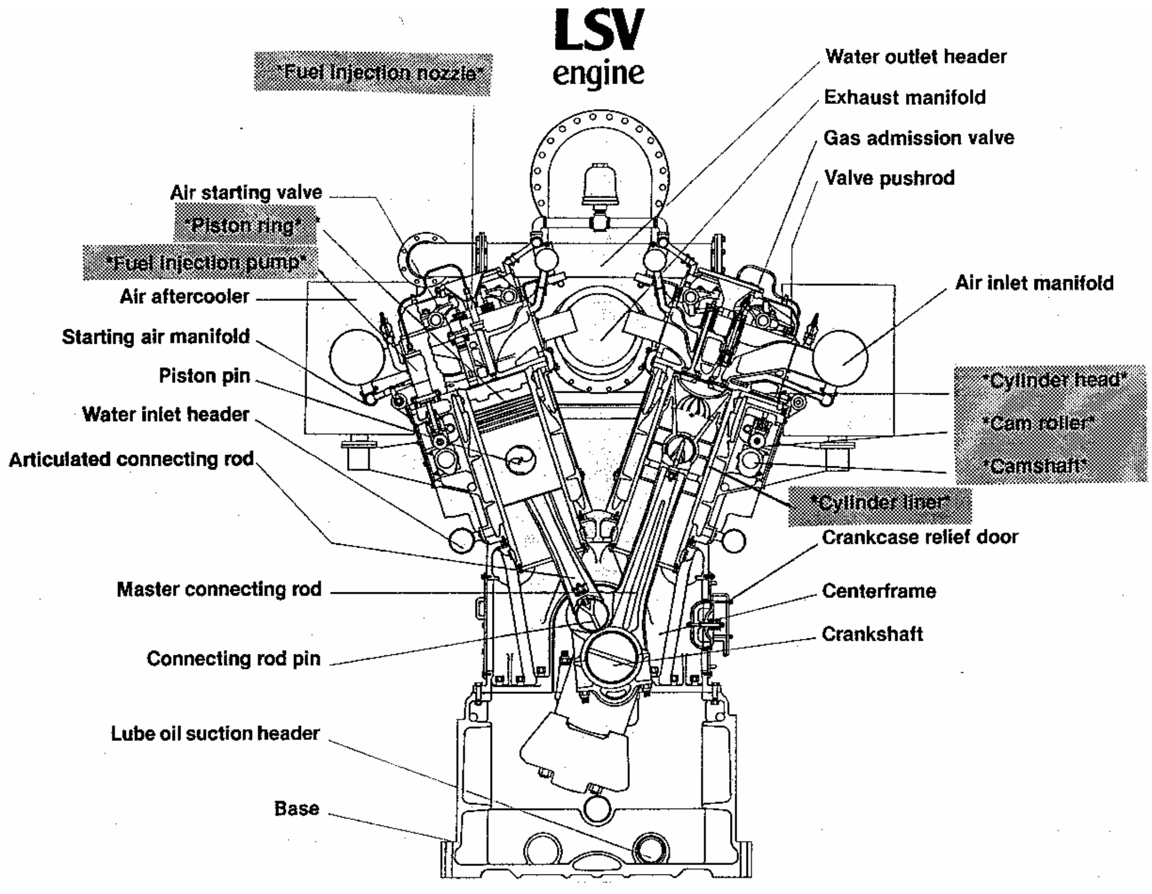


Figure 55. Highlights of Novel Coal-Fueled Diesel Engine Technology Elements

As part of this test series, Cooper evaluated the performance impact of reducing the pilot fuel quantity. A small amount of diesel fuel is normally injected into the cylinder by a separate pilot nozzle to provide a positive ignition source for the CWF fuel spray. Typically, the amount of pilot fuel has been 7 to 8% of the total fuel energy injected into the cylinder at full load conditions. Tests in February, 1996, showed that the pilot fuel can be reduced by 40% with little impact on engine performance. Peak cylinder firing pressure, indicated power, cylinder exhaust temperature, etc., all indicated that the lower pilot fuel quantity reliably ignited the CWF fuel spray. Additional testing may show that even lower pilot fuel quantity can be used which would increase the CWF / DF2 consumption ratio.

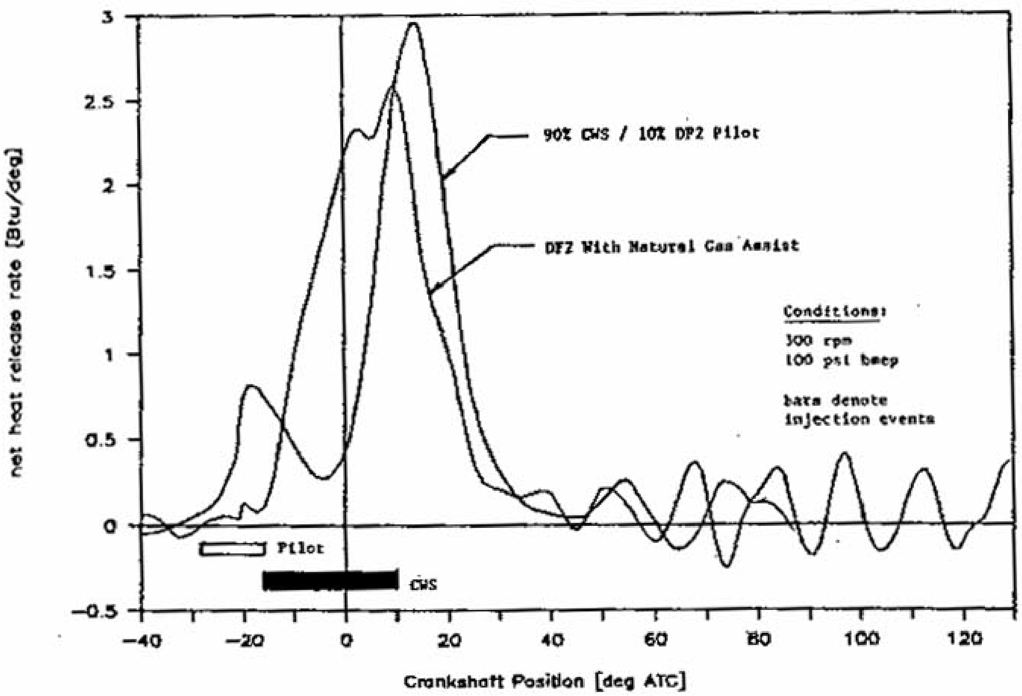


Figure 3: Heat Release Rate Curves
(DF2/NG vs. CWS)

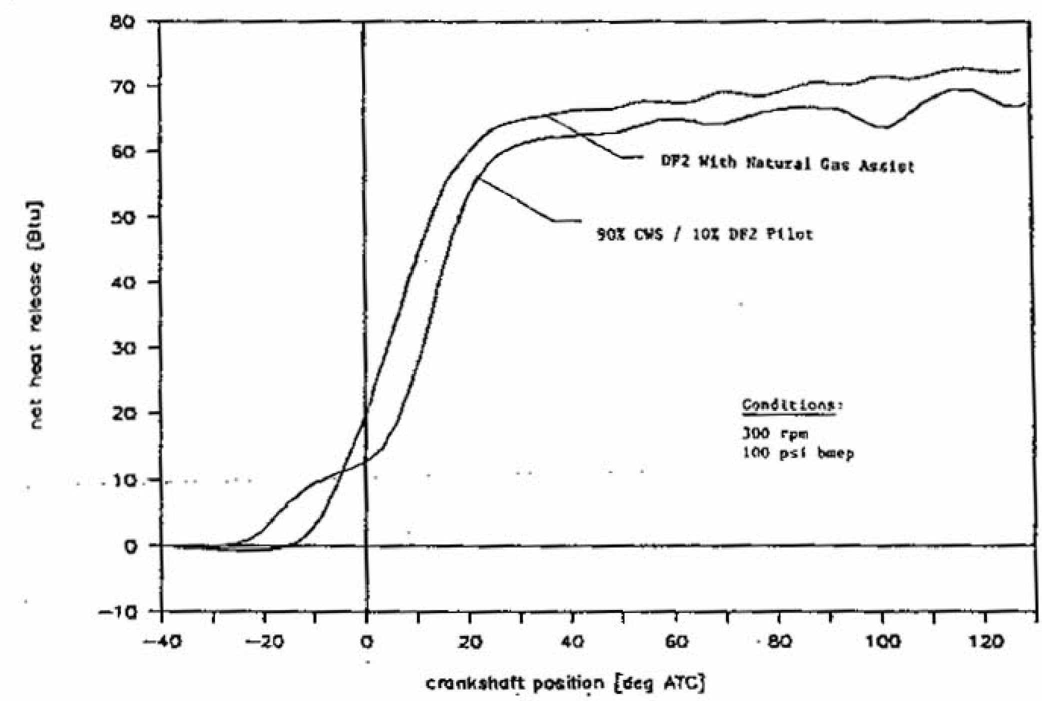
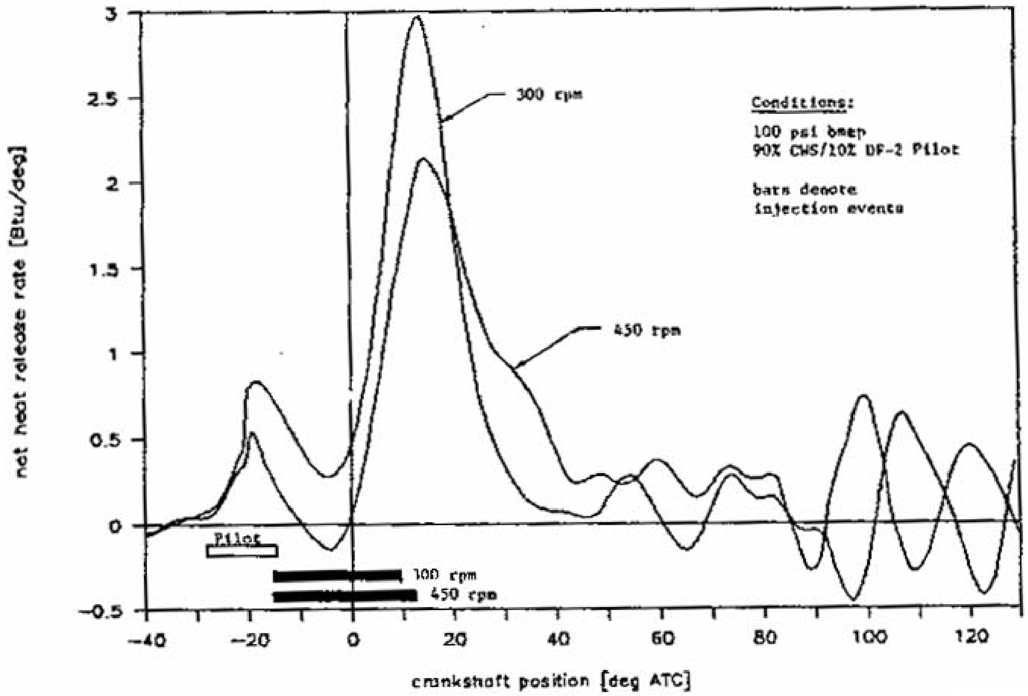


Figure 4: Cumulative Heat Release Curves
(DF2/NG vs. CWS)



net heat release rate [Btu/deg]
crankshaft position [deg ATC]
300 rpm
450 rpm
Conditions:
100 psi bmep
90% CWS/10% DF-2 Pilot
bars denote
injection events
Pilot
300 rpm
450 rpm
3
2.5
2
1.5
1
0.5
0
-0.5
-40
-20
0
20
40
60
80
100
120

Appendix D. Summary of 1970-1990 Coal-Diesel Experience

Overview

The Fossil Energy Division of the U.S. Department of Energy, through the Morgantown Energy Technology Center (METC) initiated a concentrated effort to develop coal burning diesel and gas turbine engines. The diesel engine work in the METC sponsored program was performed at Arthur D. Little (Cooper-Bessemer as subcontractor), Bartlesville Energy Technology Center (now NIPER), Caterpillar, Detroit Diesel Corporation, General Motors Corporation (Electromotive Division), General Electric, Southwest Research Institute, and various universities and other research and development organizations. This OOE-METC coal engine RD&D initiative which spanned the 1982-1993 timeframe is the topic of this review document.

In terms of total investment and technical progress, it appears that the METC Program has made more progress and successfully incorporated more diverse technologies than all of the previous efforts over the period 1890-1980 to develop coal burning reciprocating engines. At the outset of this METC program, it was clear that major technical hurdles had never been satisfactorily overcome in the previous programs, using various standard engine technologies. It was apparent that new engine component designs and new materials were needed in order to design systems that had acceptable reliability and durability. New coal fuel forms with lower ash and thus more compatible with the engine were also needed. Moreover, to meet modern environmental standards, it would be necessary to add emission control technology to make a coal engine competitive and attractive. As described in this summary, the solutions developed under the 1982-93 DOE program generally added some complexity and some additional cost to the conventional diesel engine system. However, the METC program has resulted in the development of coal-fueled engine systems that are expected to be competitive with other fossil fuels once the price of the other fuels continues to escalate.

It now appears that in the 2010-2030 timeframe, that the prices of petroleum and natural gas may ultimately increase to the point where the cost of direct coal utilization in engines will be competitive. There are no required technical breakthroughs and the coal-diesel technology now appears potentially promising for market introduction, although this must be confirmed by field demonstrations of "base-loaded" engines. Coal-fueled engine technology could be competitive in the market at that time. The accomplishments of the METC coal fueled diesel engine program must therefore be preserved for use at that time. The objective of this summary is to provide a well defined starting point for future projects in the coal fueled diesel engine area.

Organization of this Summary

First, the starting point for the METC program identified in terms of the major technical hurdles that had to be addressed. Then, the structure of the METC program to address these hurdles is briefly described and the key participants identified for future reference. The remaining sections and the major portion of the summary are devoted to discussion of the work that was done in each problem area. Emphasis is placed on the status of the work as of

1993 and identification of the best starting point for continuation of the work in the future. The last section is a summary of the key design features that must be incorporated in a successful engine, based on the current available technology. A reference list is included as well as a complete bibliography of references that are related to the development of a coal burning diesel engine.

Summary of Lessons Learned (1970-1990)

The use of coal-water slurry in medium-speed engines has been demonstrated to be technically feasible. This conclusion is based upon the use of well formulated coal-water slurries (CWS) in the range of 45-50 weight percent coal loading, using bituminous coals ground to 5-12 micron mean size particles, with a top size limit of 30-85 microns, and sulfur and ash contents in the range of 1-2 percent each by mass. The ranges in coal specification reflect the GE and ADL/Cooper engine test experience. Higher ash contents may be usable in certain larger (and slower RPM) engines, but the technology for cleaning has advanced to the point where 1-2 percent ash clean coal is reasonable to produce. Approximately 1500 hours of engine operation with CWS have been achieved in the aggregate in the GE and the Arthur D. Little/Cooper programs alone.

The current state of the technology is such that special engine and component design modifications for coal water fuels (CWS) can be very well defined for engines that operate in the speed range below 1000 rpm. The design details of modifications for CWS needed for higher speed engines are not as well defined, because the operating experience in these engines is less extensive.

The CWS fuel specification is somewhat dependent on the engine speed, in that the operating experience indicates that the slower 400 rpm engines are somewhat less sensitive to coal fuel characteristics than the 1000 rpm engine. The following specification therefore represents an attempt to satisfy the requirements of the 400-1000 rpm engines, realizing that the specified coal fuel will be marginally better than required for operation of the slower engines.

CWS Specification (125)	
Coal	Bituminous
Mean Coal Size	5 – 12 microns
Top Particle Size	15-85 Microns
Ash Content (dry coal basis)	1 – 2 Percent by Mass
Sulfur (dry coal basis)	1 – 2 Percent by Mass
Viscosity	<300 Centipoise @ 1000^{-1} sec
Additive	Dispersant

Fuel System

Details of the CWS fuel system design are described as a major portion of this summary. The following are the salient points that are common to all CWS fuel system designs and should be considered as the starting point for future work in this area:

(a) **Piping** - smooth pipe, no dead volumes, no rapid changes in flow area.

(b) **Tank** - Continuously recirculating, horizontal-cylindrical tank equipped with a floating suction, and discharge through a manifold with holes designed to produce flow with circulation.

(c) **Injection Pump** - Conventional diesel injection pump sized to inject the required amount of slurry to achieve full load and coupled to the engine, supplying diesel fuel pressure pulses to a shuttle piston assembly.

(d) **Shuttle Piston Assembly** - Conventional injection pump barrel and plunger assembly design parameters of surface finish and clearances, titanium nitride coated, sized to displace 150 percent of the required full load slurry flow. and with an L/D of approximately 1.

(e) **Nozzle Holes** - Sapphire or diamond compact inserts.

(f) **Needle Valve** - Tungsten carbide plasma coated

(g) **Needle Valve Seat -** Tungsten carbide seat insert

Combustion Chamber

The results of the METC Program indicate that conventional open chamber, direct injection, low swirl combustion chambers are acceptable for operation on CWS. Combustion efficiencies of 99 percent were routinely demonstrated in these designs if the intake air temperature and pressure are made as high as practical within the mechanical limitations of the engine components. The bowl shape can be either a shallow "Mexican Hat" design, as used in the GE and SWRI/Detroit Diesel projects, or a shallow bowl as used in the Cooper engine.

Air Temperature

It is clear that turbocharging is required to achieve acceptable breathing characteristics, especially with the higher manifold air temperatures required for operation on CWS. The manifold air temperature should be at least 135°C and the pressure should be in the range of 300kPa.

Pilot Fuel

While autoignition was demonstrated as feasible under special circumstance in the GE, the Cooper, Adiabatics, and the Detroit Diesel engines, positive ignition timing control is essential for reliable and efficient operation of a CWS engine. Pilot injection of a small quantity of diesel fuel appears to be the best method of ignition control. Pilot injection offers the opportunity for cold starting and operation at part load conditions, especially idle, where the demonstrated experience on CWS indicates that the engine temperatures drop below the levels required for reliable operation on CWS.

Rings and Liners

The rings and liners should be tungsten carbide coated and the lubricant should contain high concentrations of dispersant additive to prevent excessive wear of these components. It appears that filtration of the lube oil using the best available filter technology (pleated paper filter with 5 micron size) is sufficient to control wear rates in the rest of the engine.

INTRODUCTION

The use of coal as a fuel for diesel engines dates back to the early days of the development of the engine. Dr. Diesel envisioned his concept as a multi-fuel engine with coal a prime candidate due to the fact that it was Germany's primary domestic energy resource. It is interesting that the focus on coal burning diesel engines appears to peak about every twenty years as shortages of other energy resources increase the economic attractiveness of using coal.

This periodic interest in coal started in Germany with the work of Diesel (l)[1] in the timeframe 1898-1906. Pawlikowski carried on the work from 1916 to 1928. Two German companies commercialized the technology prior to and during World War II. The next flurry of activity occurred in the United States in the period from 1957-69, with work done at Southwest Research Institute (2), Virginia Polytechnical University (3), and Howard University (4). The current period of activity started in 1978 with work sponsored by the Conservation and Renewable Energy Branch of the U.S. Department of Energy. This work was done at Southwest Research Institute and by ThermoElectron at Sulzer Engine in Switzerland. In 1982, the Fossil Energy Branch of the U.S. Department of Energy, through what was then the Morgantown Energy Technology Center (METC), initiated a concentrated effort to develop coal burning diesel and gas turbine engines. The diesel engine work in the METC sponsored program was performed at Arthur D Little (Cooper-Bessemer as subcontractor), Bartlesville Energy Technology Center (now NIPER), Caterpillar, Detroit Diesel Corporation, General Motor Corporation (Electromotive Division), General Electric, Southwest Research Institute, and various universities and other research and development organizations. This DOE-METC coal engine RD&D initiative which spanned the 1982-1993 timeframe is the topic of this review document.

In terms of total investment and technical progress, it appears that the METC Program has made more progress and successfully incorporated more diverse technologies than all of the previous efforts over the period 1890-1980 to develop coal burning reciprocating engines. At the outset of this METC program, it was clear that major technical hurdles had never been satisfactorily overcome in the previous programs, using various standard engine technologies. It was apparent that new engine component designs and new materials were needed in order to design systems that had acceptable reliability and durability. New coal fuel forms with lower ash and thus more compatible with the engine were also needed. Moreover, to meet modern environmental standards, it would be necessary to add emission control technology to make a coal engine competitive and attractive. As described in this report, the solutions developed under the 1982-93 DOE program generally added some complexity and some additional cost to the conventional diesel engine system. However, the METC program has resulted in the development of coal-fueled engine systems that are expected to be competitive with other fossil fuels once the price of the other fuels continues to escalate.

It now appears that in the 2010-2030 timeframe, when the supplies of petroleum and natural gas recoverable at low cost begin to be limited and the prices begin to increase in response to the normal supply-demand pressures, the accomplishments of the METC Program will be of very valuable in providing lower cost coal alternatives to oil and gas. It is therefore very important that the accomplishments of this program be clearly documented, summarized, and reduced to definition of the best available technology. The initial efforts in the current

[1] All references in Appendix D refer to the list of references following the main body of the report.

objective of this report is, therefore, to provide a well defined starting point for future projects in the coal fueled diesel engine area.

The review is broken down into several sections. The next section contains a very brief summary of the historical work done before 1979. Soehngen (l), and Caton and Rosegay (5) provided excellent summaries of this work that will serve as the historical references. The starting point for the METC program will be identified in terms of the major technical hurdles that had to be addressed in each component area (fuel preparation and handling, fuel injection. ignition and combustion. wear prevention, and emissions control). The structure of the METC program will be briefly described and the key participants identified for future reference. The remaining sections and the major portion of the report will be devoted to discussion of the work that was done in each component development area. Emphasis is placed on the status of the work in 1993 and identification of the best starting point for continuation of the work in the future. The last section of the report is a summary of the key design features that must be incorporated in a successful engine, on the current available technology. A reference list is included as well as a complete bibliography of references that are related to the development of a coal burning diesel engine.

BACKGROUND

Early German Work

An excellent summary of the available historical information on coal burning diesel engines was prepared by Soehngen (l) for the Office of Fossil Energy of the Energy Research and Development Administration (ERDA), more recently the U.S. Department of Energy. This report was summarized, along with several other publications, by Caton and Rosegay (5) and presented as a technical paper for the Society of Automotive Engineers.

The history of solid fuel burning engines dates back to the work of the French scholar Montgolfier (4) in 1780. Dr. Rudolf Diesel worked periodically on the concept for approximately 10 years (1898-1908). Soehngen (l) indicated that Diesel worked on the coal diesel only because he had mentioned the use of coal in his basic engine patent. His tests were apparently limited to fumigation of coal dust into the intake air. He observed high wear and the accumulation of deposits on the piston and cylinder wall. He ceased working on the concept after an accident (possibly a coal dust explosion) occurred during operation on coal dust.

Pawlikowski, a co-worker of Diesel, apparently continued the work using dry powder coal until approximately 1928. As described by Soehngen (1), there are apparently 30 patents and 50 reports describing this work. The patents describe various concepts for introduction of the coal powder the engine and for the design of rings and seals to mitigate the wear problems associated with the use of coal. The reports have apparently been lost and were not available when Soehngen performed his review of the German work. While it may be of historical value to have access to the reports and patents, it likely that the material is of little technical value because of the low firing pressures, low efficiencies, and material and manufacturing limitations typical in that time frame.

Development of the basic Pawlikowski engine apparently continued in at least four different industrial organizations in Germany, prior to and during World War II. Soehngen (1) indicates that while these efforts may have had some qualified successful results, metered fuel injection and engine wear continued to be major problems that were not permanently overcome. It is likely that the results of these experiments would provide little new information if details were available because the key issues remained to be engine wear, fuel metering and injection, and fuel handling.

There were several major developments of this early work that have only partly been exploited in the more recent coal fueled engine program. One key observation from the older work is the fact that there were several patents developed for piston-cylinder concepts designed to prevent contact of the coal dust on the oil wetted cylinder. This was a greater problem for the early engine designs because most coal dust mixture preparation schemes did not confine the coal to a wetted spray. These experiments verify what is intuitively clear, and that is that coal particles that interact with the relatively cool lubricant film on cylinder wall will most probably not ignite and burn, and will definitely increase the cylinder liner and piston ring wear rates. Coal particles must, therefore, be prevented from interacting with the lubricant film in the engine (as much as possible). For example, fumigation of coal during the intake process, or injection of coal fuel directly on the cylinder liner would accelerate wear rates.

Another important observation from this early work is a reference to the development of hardened materials for use in the manufacture of the piston rings and cylinder liners. It appears the German company Schichau Werke developed a material and process for hardening cast iron liners and rings. The alloy composition is listed in Table 23, but no information was available regarding the hardening process. It was clear that this process mitigated the wear problem in the pre-war German engines, based on the fact that the process was sold to the other coal engine manufacturers and that there were no more references to wear problems in the remainder of the German work. Hardened rings and liners (or hard film coatings) are therefore an essential element of a commercial coal fueled diesel engine.

Table 23. German Formulation of Hardened Materials (1)

	Liner	Ring
Carbon	3-4%	2-5%
Manganese	2-7%	--
Silicon	0.5-1%	1-2%
Nickel	0.1-1%	0.5-1.5%
Chromium	--	0.3-0.7%

Coal dust fuel metering and coal dust introduction into the engines were continuing problem areas in these early engines. Low-pressure intake of coal dust through an extra valve and the use of pre-chambers were widely attempted. It appears that the most successful systems involved some form of air atomization. Erosion of internal passages, plugging and, sticking of coal-contacted moving components, and erosion and abrasion of seals and seating surfaces were problems that were encountered in all the early work. Several patents were issued in this area but no clear mechanical solution was obvious. It appears that the Germans did some fundamental work to determine the basic mechanisms of the wear problems, but they lacked the materials and the manufacturing technology required to solve these problems.

The Germans also did some fundamental combustion experiments that indicated that coal ignition and combustion rates provide the fundamental limit of the speed of operation of a coal fueled diesel. However, experiments as described in reference 1 were performed in a laminar flow system. The experiments showed that diesel fuel drops also burned at a finite rate, which limits engine speed. It is almost certain, as will be discussed in some detail in the sections describing the very recent work, that the turbulence levels in an operating diesel engine are sufficiently high that the combustion rate (coal burn out rate) in the actual engines is faster than the laminar flow experiments would suggest. Coal particle size and coal type were shown to affect the combustion rate. The early data agrees with the more recent work that indicates that coal particle size and volatility are extremely important (as will be described in later sections).

U.S. Program to 1982

Work in the United States in the period from 1945 to 1978 was very limited and consisted mainly of attempts to operate convention diesel engines on slurries of coal in diesel fuel. The most notable of these efforts were those of Marshall and Shelton (3) and Marshall and Walters (6). Additional work on coal-oil mixtures was performed at the University of North Carolina, Southwest Research Institute, and Howard University. The results of all of these efforts are of limited use because the fuels were generally not characterized terms of actual particle size and flow characteristics. Because of the recent advances in preparing coal water slurries, economic studies indicate that coal oil mixtures are much less economically viable. However, the results of the coal-oil experiments did extend the findings of the German work with dry powders. The results can be summarized as follows:

1. Fuel injection of coal-oil mixtures was a problem in conventional pump-line-nozzle systems using the standard materials and the standard clearances. Sticking of components and plugging could generally be reduced by increasing clearances, but long term durability and fuel metering continued to degrade with time.

2. Cylinder liner and ring wear was reported to be a problem in most coal-oil mixture engine experiments in which wear characteristics were examined.

3. Combustion rate and combustion efficiency were adversely affected by the presence of the coal. However, reduction in combustion performance may have been due to the effects of the coal on the flow characteristics of the fuel the resulting degraded atomization in the engine.

Two programs, sponsored by the Office of Conservation and Renewable Energy of the U.S. Department of Energy, were started in 1978 to study in detail the effects of various alternative coal fuels on operation of diesel engines. One program, performed by Thermo Electron, was focused on application in very large stationary engines. The other program, performed at Southwest Research Institute (SwRI), was focused on use of coal fuels in smaller diesel engines used in transportation. The efforts in both projects involved the use of a variety of fuels including slurries of solids in diesel fuel.

The Thermo Electron work (7) was performed in a relatively large (900mm bore) l00 RPM engine manufactured by Sulzer Brothers Ltd., of Switzerland. The fuels in the initial experiments included coal-liquids and coal-oil slurries. The experiments were extended in 1982 to include the use of coal in water slurries. Four well-characterized coal slurries with loadings of approximately 50 weight percent were studied. This represented the first detailed characterizations of these types of fuels and indicated the complex nature of the mixtures, including the non-Newtonian behavior of some of the slurries. The engine design included two complete injection systems, one for pilot injection of diesel fuel and one for the slurry. The pilot system provided absolute control of the ignition timing. A special accumulator injection system, developed on this project, apparently solved the fuel metering and injection problems encountered on all of the prior slurry experiments.

The combustion and thermal efficiencies were generally somewhat lower with the slurries, with the Otisca slurry demonstrating the same, or slightly higher thermal efficiency as compared to the baseline. Particulate emissions appeared to be higher than the baseline, but all other emissions were lower. Ring and liner wear rates were a factor of 5 greater than the corresponding rates on diesel fuel.

A limitation of the usefulness of this work was the fact that Sulzer Brothers Ltd patented the injection system design and this somewhat restricted its use in the other U.S. funded projects.

The work at Southwest Research Institute (8, 9) was performed a much higher speed single-cylinder research engine manufactured by Laboratory Equipment Company. The initial experiment consisted of screening tests of a very wide range of alternative liquids and solids, generally mixed in various concentrations in diesel fuel to provide sufficient ignition quality for auto-ignition in the test engine. The results of these experiments were presented in the form of comparisons with diesel fuel. The slurries, formulated in the baseline diesel fuel, included various biomass solids as well as coals, cokes, and carbon black. The injection system was standard pump-line-nozzle technology with increased clearances to prevent sticking and plugging.

The results of the experiments demonstrated the complexity of the rheological properties of the slurries and interactions of these properties with the injection and atomization of the slurries in the engine. A second project was initiated (l0, l1) to examine in detail the interaction of the properties of the solids, the slurry properties, the injection and atomization characteristics, and the combustion performance in research engine. The solids used in these experiments were various commercial carbon blacks and cokes that could be purchased in a range of particle sizes containing various quantities of volatiles. It was clear from this work that the maximum solids loading in the diesel fuel were limited to 30 weight percent or less due to the shear thickening nature of these mixtures. It was also clear that solid burnout was directly related to the quality of the injection process.

Morgantown Energy Technology Center Program (1982-93)

The Morgantown Energy Technology Center (METC) initiated a coal-fueled heat engine program in 1982 involving the use of coal, direct fired in both gas turbine and diesel engine systems. The initial emphasis of the program was on use of coal-water slurries in gas turbine engines, primarily because it was perceived that the continuous combustion systems in these engines would be much easier to adapt to the use of coal than the intermittent combustion in diesel engines. The ultimate successes of the diesel program and the very difficult problems associated with erosion, corrosion, and deposition in the turbine engines focused added interest on the diesels, and after 1988 the two programs became more competitive in funding levels and in their approach to commercial development.

The gas turbine program provided significant contributions to the diesel program in the way of fuel developments and emission-control technology options. These contributions will be discussed in the appropriate sections of this report. The emphasis of this report is, however, on the diesel program. It is important for future development of the coal-fueled diesel to have a clear picture of resources that were devoted to this effort in the period from 1982 to 1993. A summary of the coal fueled diesel program including the major participants of the program and a summary of the activities in each project is as follows:

The initial efforts in 1982 and 1983 consisted of laboratory experiments to explore the feasibility of the use of coal-water slurry (CWS) and coal powders in diesel engines. The results were promising, and a major procurement announcement was released by DOE-METC in 1984. Three major feasibility studies were awarded in early 1985 to three leading medium-speed engine manufacturers (Cooper-Bessemer, GE, and GM Electromotive). The goal of these feasibility studies was to identify the critical technical barriers impeding the development of the engine systems, and to assess the relative difficulty of overcoming these barriers. The focus of the Cooper-Bessemer project (Arthur D. Little as prime contractor) was the development of a stationary co-generation engine system based on the 400 rpm LS engine Model. General Electric and General Motors both focused on the development of railroad locomotive systems based on engines designed for approximately 1000 rpm. All three studies included economic assessments which identified the conditions under which coal diesel systems could be competitive in the respective applications. Over the course of the project, new technologies were developed and the economic assessments were reevaluated in terms of the new information. The results of the GE and the Arthur D. Little feasibility studies both indicated that relatively inexpensive clean coal slurries were feasible to produce. General Motors, on the other hand, concluded that the Electro- Motive engines would require the more expensive ultra-clean coals. A major DOE procurement was issued in 1988 to support proof-of-concept demonstration of full-scale coal-fired diesel systems with complete emission controls for both the GE and the Cooper/Arthur D. Little technology. Work was halted on the General Motors project because GM Electro-Motive continued the development efforts using only ultra-clean coals, which was an approach that DOE economic studies had indicated was not economically viable

The Cooper/Arthur D. Little Project was focused on the continued development and demonstration of technologies examined in the feasibility study. The goal was to demonstrate these technologies on a multi-cylinder engine complete with emission controls at the Cooper research facility in Mount Vernon, Ohio. This goal was achieved in 1993.

The GE Project also included continued development of the technologies introduced in the feasibility studies. The goal of this project was to demonstrate these technologies in a multi-cylinder GE engine operating a locomotive on the test track at the GE facility in Erie, Pennsylvania. This goal was achieved in 1992.

Selection of the medium-speed engines as the focus of the program was based on some modeling work done by Caton and Rosegay (12), which indicated that the maximum engine speed for burn-out of coal particles in the 10 micron size range was 1000 rpm. While these model results were in agreement with modeling work done later (13, 14), some preliminary experiments reported by Kakwani (l5) indicated that much higher engine speeds appeared to be possible. These results were corroborated in a project with Detroit Diesel Corporation and SwRI (16,18, 21) conducted in 1990-93 and performed a 1900-rpm Detroit Diesel 8V -149 engine used in mine haul truck applications.

A project was initiated in 1988 at Caterpillar (l7, 83, 84) to examine the use of a novel high pressure gasifier system in conjunction with a large Caterpillar engine. The scope of the project involved demonstration of operation of the engine on a gas mixture simulating the gasifier output. The project also involved demonstration of the gasifier concept that included the use of pre-loaded canisters of coal that undergoes gasification at elevated pressure.

None of the METC projects were continued after they ended in 1993, due in part to successful developments and in part to shifting METC priorities (e.g., toward natural gas advanced technology). The emphasis of U.S. Government funding for alternative fuels R&D in the 1993-2000 timeframe shifted to natural gas technology. The prices of gas appeared to be low, domestic supplies appeared to be adequate for some time, and the low carbon content appeared attractive from the standpoint of control of the greenhouse gas, CO2.

As indicated previously, the economics of clean coal engines may well become favorable in the future, as the supplies of the other energy resources are depleted or become more costly to recover. The objective of this review is to provide a clear starting point for future work in coal-fueled engines. The following sections contain detailed descriptions of the work that was done in the METC program and specifications of the fuel and engine components.

FUEL SYSTEM DEVELOPMENT

Injection system development historically had been the primary obstacle to development of coal-fired diesel engines, primarily because coal dust was fuel form rather than coal slurry. While dry powder coal was of primary interest in the early development work, it was not emphasized in the METC Program. Not only are there fuel handling and safety issues associated with the use of these finely divided powders, but micronized coal slurry technology had been significantly advanced. The emphasis of the METC Program was centered on the development of systems that could handle coal water slurries with mass loadings of approximately 50 percent coal. It was quickly realized that conventional high pressure jerk-pump injection systems could be used with coal-water slurry fuels with suitable nozzle modifications. The development of the successful injection systems was actually tied very closely with the development of the fuels. The particle loading, the particle size distribution, the coal type, and the additive package all affected the flow characteristics of the resulting slurry and, in turn, the operation and performance of the injection system. The emphasis of this section will be on describing the efforts that went into development of the mechanical components of the successful fuel systems. The resulting mechanical system specification assumes that the fuel will meet the fuel specifications discussed in a later section.

Fuel Storage and Handling

Coal-water slurries will always be unstable if the viscosity of the base liquid is kept low, and the particle loading is below the volume-filled condition, a situation where the particle loading is high enough that the entire fluid volume is essentially filled with contacting particles. In other words, if the slurry has the characteristic of a fluid of reasonable viscosity, the density difference between the coal particle and the water will always cause settling.

Coal-fueled systems must therefore incorporate special features in the fuel storage and handling systems that accommodate the separation problem. Piping systems should include smooth internal flow passages, with no rapid volume changes, or changes in flow direction.

The goal in the design of the piping system is to eliminate components that allow the formation of recirculation zones and volumes where the flow velocities become very low, and settling can occur. The piping system should also include allowances for flushing of the system with water or other clear fluid prior to shut-down.

Progressive cavity pumps have been used to accomplish recirculation in fuel tanks and for transport of the slurry from the tank to the engine (18-20). Another option is air-driven intensifier pumps. Recent work at SwRI (21) indicates that air driven intensifier pumps, typically used in high pressure paint spray systems, can be configured with components designed to handle abrasive materials. A pump in this configuration has operated reliably and at much higher discharge pressures than can be achieved using a progressive cavity pump.

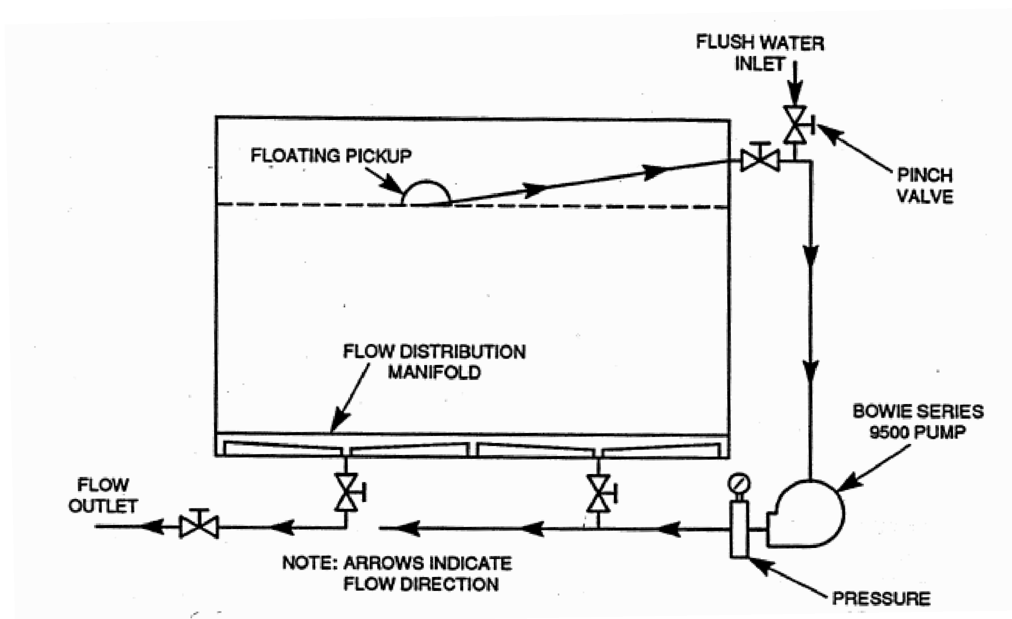


Figure 56. Coal-Water Slurry Storage Tank System

Storage of CWS in tanks presents a problem in that the coal will eventually separate if the fuel not continuously circulated throughout the tank. It is imperative, however, that the agitation be accomplished using the lowest possible shear rates to prevent "working" of the slurry and the increase in viscosity that can accompany the particle reductions that occur in "worked" slurries.

The basic concept for a storage tank system, shown schematically in Figure 56 is one of removing fuel from the top of the tank and re-introducing it at the bottom through a manifold designed to create two large counter-rotating eddies that flow outward along the entire length of the bottom of the tank and down at the center. A floating intake always insured that the pump removed slurry of the lowest concentration at the top of tank. It was felt that this was important for cyclic operation of the system and for those situations where the circulation system was shut-off for an extended period of time. In the worst case of total settling, the pump removes the water layer on the top of the tank and re-introduces it at the bottom, where its kinetic energy is used to re-mix the slurry.

Computational fluid dynamics was used to design the manifold located at the bottom of the tank and to the appropriate flow velocities, and in turn to design the piping system and select the pump. In non-dimensional form, the optimum design condition is one in which RexH=5.7, where the Reynolds Number (Re) is based on the tank diameter and the flow velocity at the exit of the manifold, and H is the ratio of the manifold discharge orifice diameter to the tank diameter. A tank system designed to these specifications was used to store a slurry of 51 weight percent coal for a two year period at SwRI. The system was operated on a 15 minutes on/off cycle over that period. Experience with a rotating vane pump and progressive cavity pumps indicate that these are good choices. Intensifier pump designs may also be a good choice, depending on the pressure requirement of the fuel system. The Cooper/ADL project had similar coal slurry storage for two years in a 6,000-gallon tank with periodic recirculation.

Injection System Design

Diesel fuel injection systems have three functions: (1) to meter the quantity of fuel needed to maintain the engine speed and load, (2) to inject the fuel at high velocity into the combustion chamber, and (3) to time the start of injection. Older designs include an injection pump that is coupled directly to an output shaft on the engine. The pump simultaneously provides the fuel metering and high pressure pumping functions. The high pressure pulse of fuel is supplied through a length of tubing to an injection nozzle that incorporates a needle valve that opens in response to the fuel injection pulse. Fuel passes through the seating area of the needle valve, typically forming a spray by radially flowing through several small diameter holes and into the engine. The goals in designing the pumping element and the nozzle hole configurations are to provide sufficient turn-down to operate the engine from idle through full load, and to provide sufficient flow velocity in the fuel jets so that they penetrate across the combustion chamber, in order to utilize all the air. Impingement on the combustion chamber walls occurs in some design with satisfactory results.

Some current designs for highway truck diesel applications are called unit injectors and they incorporate the pumping and metering functions inside the injection nozzle. New unit injectors also incorporate an electrically actuated solenoid valve that is used to accurately control the beginning and ending of injection over the entire speed-load range of the engine.

The manufacturing technology for injection systems has evolved to the point where tolerances in the range of 2.5 to 25 microns are common in the barrel and plunger assemblies and the needle valves of these systems. These clearances are smaller than the mean particle size of most slurries, but the size distributions are typically Gaussian and contain a sufficient number of very small particles that can enter the clearance and cause wear and sticking of the elements. In addition, finite element analysis of the barrel assembly performed at SwRI as a part of the work described in Reference 22[2], indicated that the barrel expands sufficiently during injection to admit particles that are significantly larger than the static clearances. These theoretical considerations were verified in several experiments as part of the feasibility projects, where standard injection systems were used for testing slurries. Typical times to failure for these tests ranged from instantly, to a few minutes. It was observed, however, that relatively long periods of operation could be achieved if the tolerances were actually increased and the slurry was allowed to leak through the sealing areas (18). It should be noted that failures, in the form of increased leakage and decreased fuel delivery, always

[2] List of references following main body of report.

occurred with these systems, even when sticking was not a problem.

Several different slurry injection system designs were evaluated in the various projects of the feasibility study. The work at SwRI included examination of two different pump-line-nozzle systems with increased clearances (23,24), a unit injector with increased clearance (24), a Cummins Pressure- Time system (25), a unit injector equipped with a diaphragm to separate diesel fuel and slurry inside the injector, and a pump-line nozzle system equipped with a free piston located between the nozzle and the injection, and separating the slurry from diesel fuel that is circulated through the injection pump (21) . The initial efforts at GE (26) also involved the use of a diaphragm system, as well as an accumulator system that used the diaphragm system as the pressurization system for the accumulator (27). The basic layout of the GE diaphragm is shown in Figure 57 and a cross-section of the accumulator nozzle is shown Figure 58. The Arthur D. Little-Cooper-AMBAC team worked on the development of a pump-line-nozzle system that incorporated a free piston (called by AMBAC a shuttle piston) to separate the two fuels rather than a diaphragm (28), as well as a common rail system that was not pursued once the shuttle piston system proved reliable. A shuttle piston system was also developed at SwRI for use in the High-Speed engine tests on the DDC 8V-149 engine.(21)

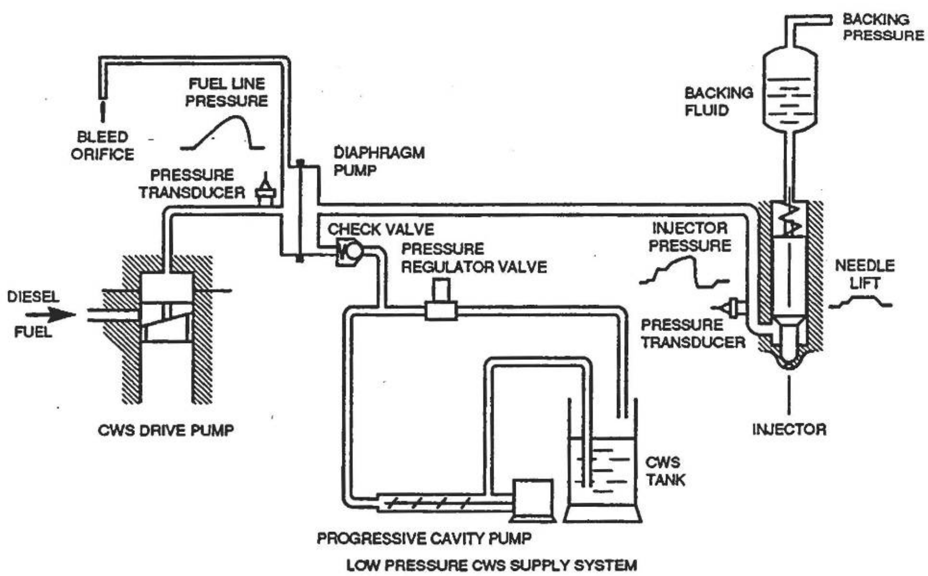


Figure 57. General Electric Fuel System Schematic

It became clear from test results that it is essential that the slurry be separated physically from the barrel and plunger assemblies of the injection system. Both the diaphragm concept and the shuttle-piston concept accomplish this objective. Other solutions such as increased clearances in conventional injection systems, and conventional surface preparation did not prove to be acceptable.

The shuttle-piston approach proved to be the "winner" because the design constraints on the diaphragm systems tested in both by both GE and SwRI proved to be unacceptable. Fatigue failure was a problem with diaphragm systems, almost independent of the material selection, if the diaphragm was made small enough to produce acceptable injection pressure characteristics. Increasing the size of the diaphragm to reduce the magnitude of the deflections resulted in a large increase in the volume of the high pressure section of the system the GE Team (30) abandoned the diaphragm approach in favor of the shuttle piston concept.

The shuttle piston approach developed by the Cooper/Arthur D. Little Team (with AMBAC) has evolved as the most reliable CWS injection system concept. Basically, the injection system looks like a conventional pump-line-nozzle system, except that a free piston, or shuttle piston assemble is installed in the injection line between the pump and the nozzle. The section between the injection pump and the shuttle piston is filled with diesel fuel that acts on one side of the shuttle during the injection event. Motion of the shuttle translates the pressure pulse to the slurry that fills the other side of the system. The pressurized slurry then opens the needle valve in the injection nozzle and slurry is injected into the engine. Development of this technology and that used in the Arthur D. Little Project was done with significant guidance from Mr. Jack Kimberley of AMBAC. The designs developed in the other projects share the same main features.

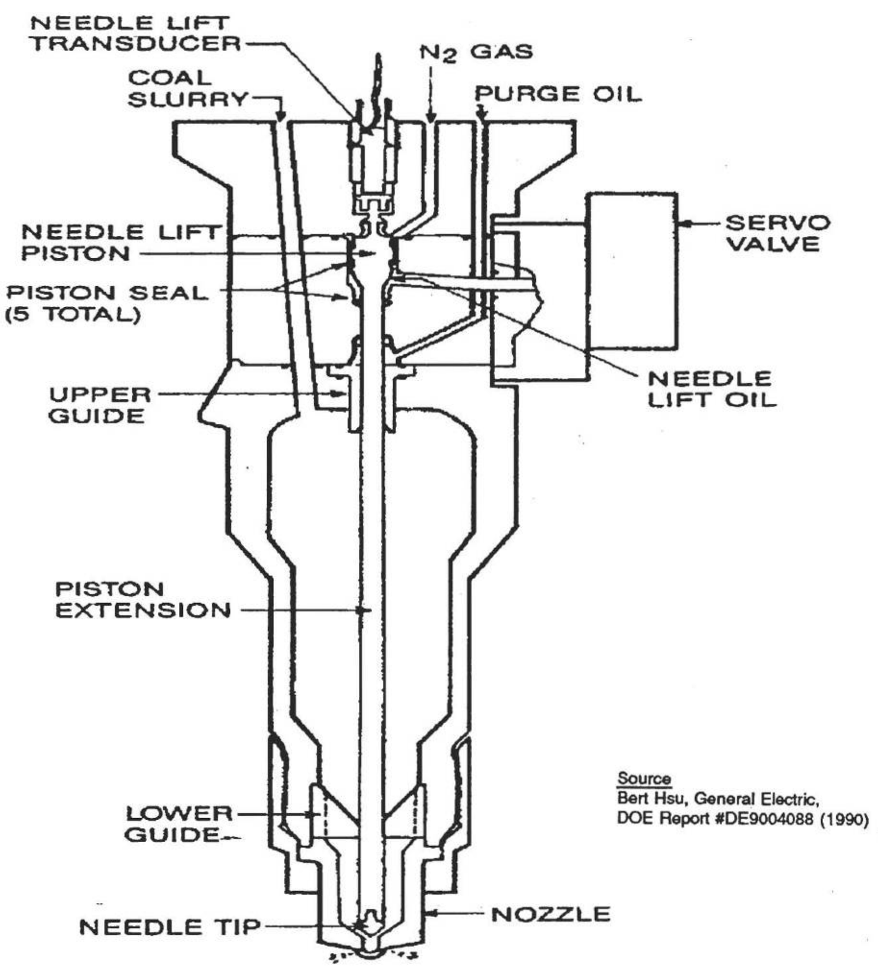


Figure 58. Schematic of the General Electric Accumulator Injection Nozzle

The design details of the shuttle piston assembly are very similar to those used in standard injection system barrel and plunder designs, in terms of the clearances (on the order of 2.5 to 25 microns clearance on the diameter) and the surface finish. The displacement volume of the shuttle must be approximately 50 percent larger than the fuel delivery required for full load. This excess capacity insures that the shuttle will not “bottom out" and will accommodate those situations where the shuttle does not return completely. Durability of the shuttle piston assembly is greatly improved if the shuttle is coated with titanium nitride and the barre1 made of nitralloy. The GE Team developed a shuttle piston design in which the piston was made from tungsten carbide. Durability of this system was also acceptable.(31,85,86)

Selection of the shuttle diameter and stroke is governed by the response time required for return of the shuttle and the compressibility of the trapped fuel. The longer the stroke, the longer the return time. The larger the diameter, the larger the trapped volume by the square of the diameter. It appears that a good rule of thumb is that the diameter should equal the diameter of plunger in the injection pump, and the length of the stroke should be as close to the diameter as allowed by the 50 percent excess volume requirement.

Seal oil ports supply seal oil to the center of the shuttle piston stroke and to the center of the guide area of the needle valve. The seal oil is usually supplied using intensifier pumps operating with discharge pressures greater than the peak injection pressures. The seal oil acts to lubricate the moving parts and prevent the passage of coal particles into the tight clearances. The experience at SwRI (21) and in the Arthur D. Little work (32) indicated that the seal oil must be supplied to both locations. Optimum operation of the shuttle piston is accomplished if the piston is undercut in the region where the seal oil is introduced. Experience at SwRI indicated that the method did not work as well if the flow clearance is made in the barrel assembly.

The shuttle piston designs developed at Cooper/Arthur D. Little, SwRI, and GE all emphasized the use of nearly conventional injection nozzles. The GE Team did some preliminary development work on an accumulator design that was intended to be used in their project on full scale locomotive demonstration of the system. (26) Single-cylinder engine experiments using the GE accumulator injection nozzle were very promising because the design offered the opportunity for variation and control of the injection timing. (30) A schematic of the system, reproduced from Reference 30, is presented in Figure 58. The accumulator housing in the injection nozzle was pressurized with slurry to injection pressure using a shuttle piston injection system. Injection timing was controlled by an electrically actuated solenoid valve that allowed high pressure hydraulic fluid to open the needle valve.

Injection System Materials

As indicated above, wear prevention in the shuttle piston assembly was greatly improved by the appropriate selection of materials and coatings. Two other significant areas for wear in the injection system are the nozzle holes and the needle valve seat.

Nozzle Holes

Dunlay et al. (33) reported significant nozzle hole wear in the Sulzer engine with conventional steel nozzles after a few hours operation on a 34 percent coal in oil slurry. Nydick (34) reported nozzle hole enlargement sintered tungsten carbide inserts in the same Sulzer engine. Based on results obtained from four different slurries, Nydick concluded that the wear was more dependent on particle size than ash content. Ryan et al. (25) reported a doubling of the nozzle hole diameter after 25 hours operation on a 50 percent CWS during bench tests using a conventional Cummins P- T nozzle made from carbonized steel. Hsu reported 10 percent decreases in the injection pressure after minutes of operation on CWS using conventional carbonized steel injection nozzle tips. Schwalb et al. (21) reported wear rates of from 0.5 to 1.6 percent increase in the nozzle hole flow area per minute run time, using conventional steel alloys. Rao et al. (28) reported increases in the nozzle hole exit diameter of up to 50 percent after less than one hour of operations on low ash (less than 2 percent) coal, and substantially more wear with a 3.8 percent ash coal, a conclusion that is in direct conflict with Nydick's (34) conclusion regarding the importance of ash content.

Hard steel alloys, coatings of very hard materials, and monolithic ceramic and hardened materials have been tested in both bench experiments and in actual CWS engine tests. Some caution must be exercised in drawing absolute conclusions from the results of bench experiments. Both the Cooper/Arthur D. Little (36) and the GE (31) Teams reported significant nozzle hole wear in conventional materials during engine tests with CWS. Both Teams reported a wear pattern in which the entrance to the hole showed abrasive rounding, the exit showed enlargement that greatly exceeded the entrance enlargement, and wear pattern in the hole with a "trumpeting" appearance (see Figure 59, reproduced from Reference 37).

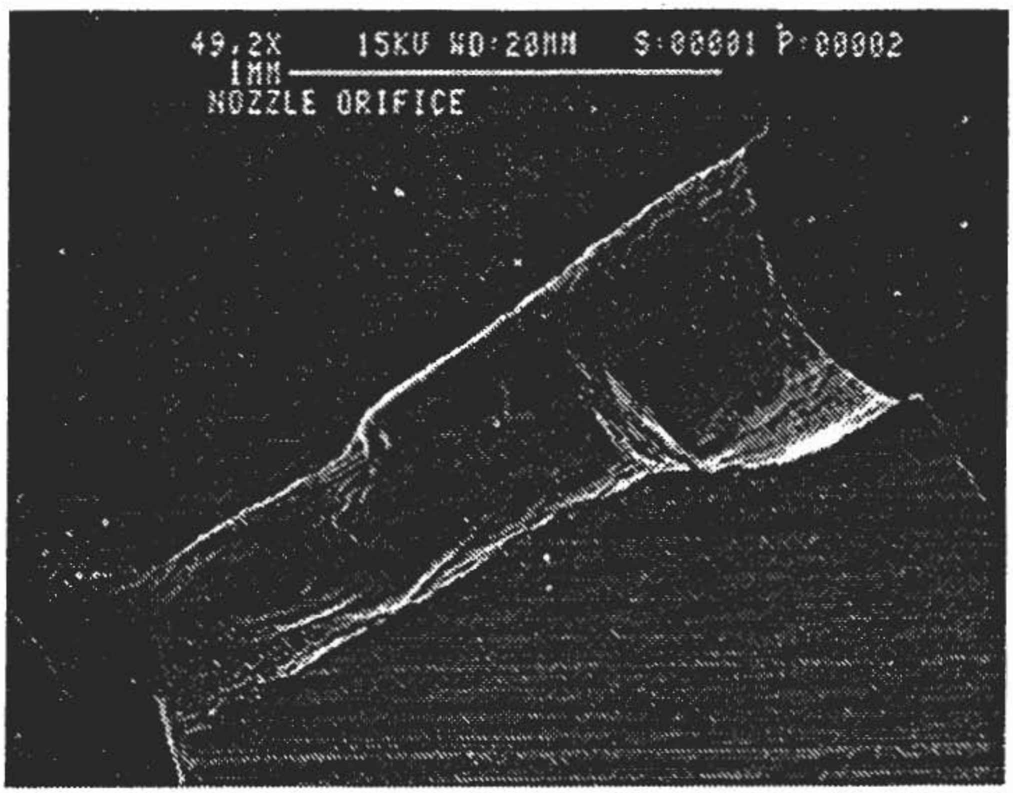


Figure 59. Injection Nozzle Hole Wear Pattern

Efforts to reproduce these wear patterns in continuous flow bench experiments were unsuccessful at Arthur D. Little and SwRI (38) and at GE (31). The bench experiments were all performed using continuous flow through an orifice assembly in which the materials and the configuration of the hole could be easily changed The two differences between these experiments and the actual engine were continuous versus transient operation, and the temperature. It was suggested that the trumpeted wear was due to cavitation of the CWS, starting at the exit to the nozzle. It was hypothesized that the coal particles the slurry provided high density solids that enhanced the wear during collapse of the cavitation bubbles. Temperatures in the test fixtures presumably were too low to support cavitation. The hard materials that were tested in all of the studies as a means to reduce the wear rate are listed in Table 24. The ranking of the materials are also listed as reported by the different project teams. The GE rankings presented in the table are based on the results of bench experiments.(31) Another example of the problem of using bench experiments is the fact that engine experiments at GE using the Kennametal K313 tungsten carbide indicated high wear rates. The Cooper/Arthur D. Little data were all obtained during actual engine tests.

Despite the variation in results in Table 24, it is possible to select some of the more promising hard materials. It is generally agreed that the best material choices are selected formulations of tungsten carbide/cobalt, cubic boron nitride, diamond compact, and sapphire.

Table 24. Material Evaluated for Use in Injection Nozzle Holes

Materials	Relative Wear Rates		
	Cooper/ Arthur D. Little	General Electric	DDC/SwRI
Steel Alloys	High	High	High
Titanium Nitride Coating	High	--	--
Stellite 6B	--	High	--
Titanium Alloy/Titanium Diboride Coating	High	Low	--
Carboloy 883 (Tungsten Carbide/6% Cobalt Binder	--	Moderate	--
Tunsten Carbide 6% Cobalt Binder (Kennamental K313)	--	Low	--
Tungsten Carbide/Cobalt Binder (Kennamental K714)	--	--	--
Cubic Boron Nitride	--	Low	--
Silicon Carbide (Hexalloy SA)	--	Moderate	--
Diamond Compact	Low	Low	--
Sapphire	Low	Low	--

It appears that the performance of the tungsten carbides depends heavily on the formulation, with the finer grain carbides demonstrating lower wear rates. The Cooper/Arthur D. Little Team (36) reported very low wears rates using monolithic assemblies fabricated from the Kennametal K714 formulation. The assembly consisted of a piece, called a button, that contained the nozzle holes and that was held in place at the tip of the nozzle by compression, using a screw on retaining nut. Based on observed wear rates after 150-200 hours of testing, it was projected that these components would survive from 500-2000 hours of engine operation on CWS. The GE team concluded that the tungsten carbide family of materials does not have sufficient erosion resistance for use in the nozzle holes.(39) The GE team did, however, propose a similar approach to installing very hard materials in the location of the nozzle tip holes, with the "button" actually fabricated as a small insert that is installed inside the, tip in the region of the holes.

Experience with the cubic boron nitride have been excellent, but the results with the diamond compact material and the sapphire have been as good.(40) The diamond compact and the sapphire offer a significant cost advantage over the ceramics in the fact that they are both used commercially in the configuration needed for use in the nozzle holes. The diamond compact is a composite that is commonly used in wire making dies. Commercial dies are available with holes of dimensions typical of diesel injection nozzle holes. Sapphires are available commercially as jewel bearings for mechanical chronometers and fine motors. They are also available with holes of the appropriate dimensions. Both the sapphires and the diamond compact dies are available in cylindrical geometries with the orifice running through the axis.

Both the Cooper/Arthur D. Little and the GE teams have developed designs in which these small cylindrical pieces are installed as inserts in the injection nozzle tip in the locations of each of the nozzle holes. The diamond compact inserts, a product of GE Specialty Materials, are brazed in a stainless steel nozzle tip using very precise control of the brazing temperature to prevent graphitization of the diamond. (39) The sapphires are swaged into a small metal support that in turn is electron beam welded into the nozzle tip. (36)

Needle Valve Seats

Operation of conventional nozzles on CWS results in wear in two areas of the needle valve (36-39), even with seal oil supplied to the guide section of the needle. The leading edge of the guide section of the valve becomes rounded after approximately 150 hours of operation. The other area of wear is on both elements of the valve seat. The conclusion from both projects is that the seat should be a tungsten carbide insert used in combination with a tungsten carbide plasma coated needle valve. Not enough operating time was accumulated to determine if the barrel (above the seat) also needs protection.

COMBUSTION SYSTEM DEVELOPMENT

Combustion of CWS involves several very complex processes that are occurring in the combustion chamber once every engine cycle. The event begins with injection of the non-Newtonian, two-phase fuel at pressures that can exceed l00MPa, into an environment of high-temperature, high-pressure air and combustion products. The fuel jets, traveling with initial exit velocities up to 300 m/sec, entrain air and slow to 20-50 m/sec, and then traverse approximate 7.5 to 20 cm prior to impacting in the combustion chamber. Conventional wisdom suggests that the jets entrain and mix with air during the 1-10 milliseconds that is available before they impact on cold combustion chamber (piston) walls. Ignition of CWS is generally thought to involve a three-phase process in which (1) the water evaporates, (2) devolatization of the coal occurs, and (3) ignition of vapor mixture occurs. Theory also suggests that combustion of the volatiles then occurs at the edge of the fuel spray, followed by char combustion via carbon oxidation to CO on the surface and continued oxidation to CO2 in regions surrounding the char particles. The location of initial ignition of the spray is obviously farther from the nozzle tip than when there is already a standing flame into which the remaining spray is injected.

Estimates of the characteristic time of the overall mechanism of coal combustion in engines suggests that CWS can be used only in medium-speed engines (e.g., 30 micron coal particles can burn out within 40-50 crank angles at 1000 rpm. The results of both Kakwani et al.(15)

and Schwalb et al.(2l), however, show that engine speeds up to 1900 rpm can be achieved with acceptable coal combustion efficiencies, provided that most of the coal particles are approximately 20 microns or less. One difference between the theoretical and laboratory measured combustion rates and those observed in engines is the higher turbulence levels that are present in engines (which might contribute to higher coal burn rates).

As will be discussed below, the fuels are non-Newtonian and can shear-thicken during injection, and possibly exhibit poor atomization characteristics. As the apparent viscosity increases, the air entrainment rate can decrease with an accompanying increase in the jet tip velocity, and increasing the probability of jet-wall interactions. Impingement of the unreacted air/slurry plume on the solid piston surface enhances the mixing process through deflected plume mixing and entrainment. Deposition of the coal on the relatively cool (500-800 F) combustion chamber walls may occur temporarily, but Cooper observed no accumulated layer on the piston after over 200 hours of operation. Coal type, source, processing, and particle size distribution all contribute to the rheological properties of the CWS. In addition, these properties control the ignition and combustion characteristics of the CWS both directly, through the volatile content and composition, and indirectly, through the atomization characteristics.

Direct impingement of CWS sprays on lubricant wetted surfaces (cylinder walls) is to be avoided because it would lead to deposition of the unreacted coal on the surfaces, degraded combustion efficiency, and greatly increased cylinder liner and ring wear. Based on test results, it appears that the CWS can impinge on the piston crown with satisfactory atomization and ignition characteristics. It is believed, however, that certain configurations of engine designs and long duration CWS spray events used in the METC Program may have resulted in some fuel impingement on the cylinder liner. Certain engine designs incorporated very shallow combustion chambers that may have allowed some fuel splash and direct impingement on the liners, depending on the injection timing and the injection rate. The piston "uncovers" the liner after several tens of degrees crank angle after IDC. In addition, suspended residual traces of ash and unburned coal are available in the combustion chamber and no doubt contributed to the ring and liner wear, even if the combustion efficiency was nearly 100 percent. This wear effect is believed to be greater when the coal/ash particles are smaller than the diametrical clearance (piston liner).

The combustion system development work is discussed in terms of the basic coal combustion considerations important to operation of CWS in an engine and the actual engine development work. Each of these items is discussed in the following sections.

Basic Combustion Considerations

In addition to the injection studies, the ignition characteristics of dry coal and coal slurries have also been examined. Clingenpeel, et al. (41) in experiments with slurries of 45 wt% coal and non-petroleum carriers, found that an engine run without a pilot injector and a glow plug would suffer a torque loss of approximately 20 percent as compared to an engine run using combustion assist mechanisms. It was also noted by Robben, et al. (42) that, without combustion assist devices, significantly higher compression temperatures (1100 to 1200K) would be required. Kamo, et al. (43) conducted tests in which he coal powder with a maximum particle size of 20 microns directly into the intake air manifold of a single-cylinder Caterpillar engine. The engine was modified to an adiabatic design by coating the combustion chamber with a ceramic coating and removing the water cooling system. These changes brought about a 40 percent reduction in heat rejection by the engine. They found that powdered coal could be run in the engine without the aid of a diesel fuel pilot. Exhaust unburned coal was not measured, however.

Other researchers of coal ignition have concentrated on studying the ignition delay time. Murdoch, et al.(44) studied the ignition delay of coal/water slurries and found it to be a function of water content, initial temperature, coal type, and particle size. They noted that the ignition delay decreases with preheating of the slurry and with decreases in particle size, and that ignition delay increases with the water content of the slurry. This increase is caused by the extra time required to vaporize the water in the drop and expose the coal particle. Siebers and Dyer (45) conducted tests in a constant-volume combustion bomb in which they compared the ignition delay of a 46 wt.% coal-water slurry to diesel fuel (DF-2). They determined that the ignition delay for the slurry was temperature and pressure dependent and increased by a factor of 5 over that of DF-2 fuel

The combustion of coal is another area that has undergone a great deal of study. Research has focused on basic combustion topics, such as particle reaction rate, and more applied topics, such as defining required temperatures and pressures for coal slurry combustion in engine-type environments. Sakai and Saito (46) concluded that the combustion of coal slurry fuels is a two-stage process consisting of gas-phase combustion followed by solid combustion. The first stage, or physical stage, is for the vaporization of the carrier. The second stage, or chemical stage, is the substantially longer period that is required for the coal agglomerate to react. Szekely and Faeth (47) found that the reaction of the agglomerate required 90 to 95 percent of the lifetime of the particle, even where agglomerate reaction rates were at a maximum. They added that the introduction of a catalyst increased agglomerate reaction rates, except at diffusion-controlled conditions. Wells, et al.(48) studied the chemical stage at various temperatures by looking at the char reactivity. They determined that at low and medium temperatures (550 to 750K and 1070 to 1270K, respectively) char reactivity was dominated by parent coal type. However, they noted that char reactivity was dominated by the pore diffusional resistance of the agglomerate at high temperatures (1300 to 1700K). Liu, et al. (49) conducted experiments in which they studied high-intensity pulverized coal flames with water injection. They found that the water injection did not affect the coal reactivity, although it did lower the flame temperature. However, they also demonstrated that "the controlling reaction mechanism through the whole temperature range is physical (diffusion), not chemical (kinetic)," this being in direct contrast to coal combustion behavior in normal intensity flames. Both Law, et al.(50) and McHale, et al.(51) in studies of coal/oil mixtures for furnaces and boilers (typical droplet sizes 100-200 microns) found that internal circulation, or swirl of the secondary air is an important factor in combustion. They reported that a high degree of swirl resulted in a significant improvement in carbon burnout. If true for the much smaller drop in diesels (20 microns), this would lead to the hypothesis that increased turbulence within the cylinder would similarly improve coal particle burnout in a diesel engine. However, droplets in the 20 micron size range quickly equilibrate to local flow velocities even in turbulent flow.

The time required for a particle of coal of a radius to burn completely is known as the coal-burning time. The faster the coal burns, the higher the allowable engine speed. Some of the first studies of coal-burning times were conducted by Nusselt and Wentzel (l). They performed tests on pulverized coal in pressurized combustion vessels to simulate engine conditions. They concluded that coal particles in the 80-100 micron size range were slow burning and that the maximum engine speed for which coal dust would be practical as a fuel was 400 RPM. As a result of their findings, most of the early coal-dust-fueled engines were designed for low-speed and low-power operation. It was not until later, when performing similar tests on diesel oil, that it was realized their combustion vessel produced burning times much longer than those in an actual engine due to the combustion vessel's lack of turbulent conditions that exist in the cylinder of an engine.

Essenhigh (52) studied the burning times for ideal spherical particles in dust flames. He based his study on two different means of reaction control: diffusion and chemical reaction. He noted that when both mechanisms were present in particle burning, the effects of two were additive. He also realized that turbulence could reduce burning times, but felt the decrease would be 50 percent at most, depending on particle size. Essenhigh and Yorke (53) conducted tests on the effects of coal particle shape and swelling on burning times. They determined that shape had no significant influence on burning times, because irregularly shaped particles burn down to a spherical shape by the time they are half burned. They also found that coal particles swell according to coal rank in two stages to a final diameter that is 1.5 times larger than the original diameter. When this swelling was taken into account, the values they measured for burning times corresponded to those calculated using the Nusselt square law. Both Essenhigh (52) and Kanury (54) presented combustion time data for coal and carbon particles. These data are summarized in Figure 60 (43). Typical ideal combustion duration for engines is 30 degrees of engine crankshaft rotation at rated power (more than 30 degrees and efficiency suffers). Based on this criteria, engine speed is plotted along the right axis. The results in Figure 60 indicate that the size of the dry particles must be in the range of 5 to 50 microns for engine speeds from 400-2000 rpm.

Rawlins, et al. (55) also studied coal particle burnout rates. They performed their tests in a vertical laboratory-scale combustor on coal/water mixtures of 70 and 73 wt.% coal. The mass-mean coal particle diameter of the slurry was about 20 microns, although due to swelling and agglomeration, some of the particles reached 200 microns in a diameter. They found that with secondary swirl the coal burnout was around 98 percent for average particle residence times less than 100 ms (equivalent to 30 deg crank angle at 100 RPM). Holve (56) used a numerical model to determine that agglomeration increases burning time by a factor of four to eight. His model also predicted that particles less than 30 microns should react completely in less than 35 milliseconds (equivalent to 30 deg CA at 300 RPM), while those less than 10 microns would take than 8 ms to burn out (equivalent to 30 deg CA at 1000 RPM). These values are similar to those obtained by Essenhigh (52). Petela (57) used a mathematical model to analyze the combustion of a coal/oil droplet. He noted that the combustion of the oil part of a coal/oil droplet takes less time due to the larger surface area of the oil and went on to predict that "if small enough coal particles were used in the coal/oil mixture, and if agglomeration could be prevented, overall combustion time of a coal/oil mixture droplet can be shorter than that of a pure oil droplet of the same size." Kamo, et al. (43) also predicted that the burning time of micronized coal dust should be short enough to allow engine speeds up to 2000 RPM.

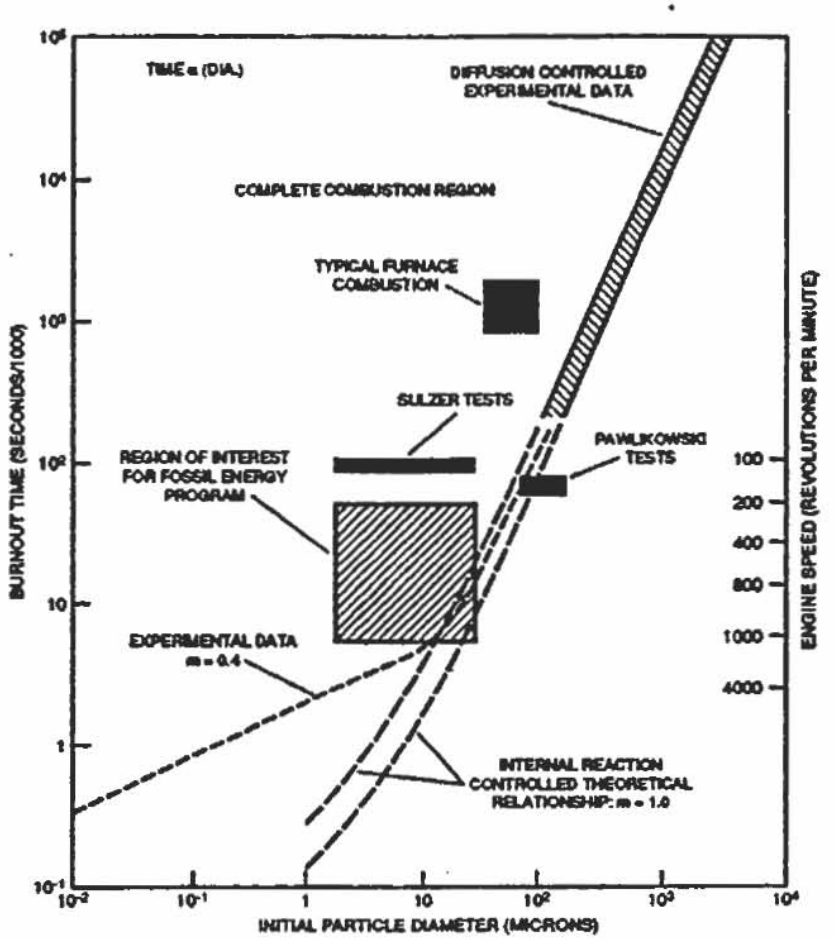


Figure 60. Coal and Carbon Burnout Time and Engine Speed versus Particle Size

The size of the coal particles in the slurry may also have an effect on how well combustion takes place, depending on whether the slurry droplets (clusters of coal particles) disintegrate upon water evaporation. Caton (58) used an engine cycle simulation based on a thermodynamic analysis with a particle combustion model to study the combustion characteristics of coal in a reciprocating, locomotive engine. He found that engine efficiency decreased with increasing engine speed. However, the decrease was for smaller particles, and negligible with particle diameters less than 20 microns for engine speeds of 250 to 1000 RPM. This finding is approximately consistent with the GE engine test data if one assumes that the slurry disintegrates into coal powder. Fu, et al.(59) performed tests in an oil-designed l00-hp fire-tube boiler to determine the effects of coal particle size on combustion properties of coal/water fuels. They found that the use of micronized coal/water slurries did not provide a significant increase in carbon burnout or boiler efficiency compared to coal/water slurries made with larger coal particles. However, available combustion residence times for boilers are several hundred milliseconds, so one would not expect to see any effect of particle size.

Other studies have focused on what conditions are required for effective combustion in an engine-type environment. Siebers and Dyer (45) conducted research comparing a coal-water slurry with DF-2 in a constant-volume combustion bomb. The slurry was made up of 45.8 wt. % low-ash coal with a maximum particle size of 22 microns. They simulated diesel engine conditions in the bomb by igniting a premixed charge of hydrogen, oxygen, and nitrogen. They found that the rate of energy release during the coal combustion was slower for the slurry than for DF-2. They also determined that the minimum temperature in the bomb at which a significant portion of the slurry burned was 800K, Leonard and Fiske (26,60) also performed experiments with coal/water slurries in a combustion bomb, using the ignition of a premixed charge to simulate the conditions in a diesel engine cylinder. However, they used higher injection pressure and gas density order to best approximate combustion in a medium-speed diesel engine at full power. They found that after ignition the coal-water slurry burned faster than diesel at initial gas temperatures above 850K. At lower temperatures, the combustion quality rapidly deteriorated. These results are clearly shown in Figure 61 where the percent burned drops very quickly as the initial temperature approaches 850-900 K. They theorized that below 850K, the cooling around the particle due to water evaporation inhibited complete combustion of the coal. They also felt that some of the unburned coal in their tests was due to the combustion bomb wall temperature being only 300K, as opposed to 500K for the walls and piston in an engine. They felt this temperature difference would decrease the probability that the coal that impinged on the wall would burn.

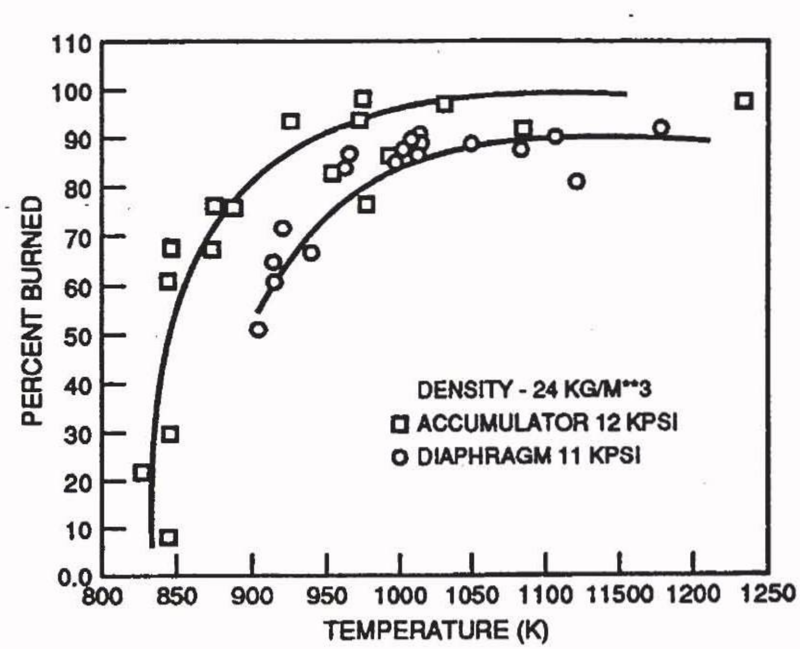


Figure 61. General Electric Coal Water Slurry Bum Rate Data

The conclusions that can be drawn from these basic considerations of coal combustion are that the slurry properties must be tailored to meet the atomization and combustion requirements imposed by the time available and the thermodynamic conditions of the engine environment. The mean particle size of the coal in the slurry should probably be in the range of 10 microns mean with a loading on the order of 50 percent by weight, and 95% of particles less than 30 microns. Particles larger than these take too long to burn at 1000 RPM, and particles much smaller than these are too expensive to produce. The compression temperature must be as high as possible in order to achieve ignition and acceptable combustion rates. In fact, pilot injection of diesel fuel was suggested by the ignition temperature and combustion rate measurements in the basic experiments. The validity of these conclusions in the engine environment are discussed in the following section.

Engine Development

Coal-water slurries in the range of 50 percent by weight coal loading have been successfully tested in ten different engines during the METC Coal Diesel Development Program. Nelson (61) performed experiments in a single-cylinder direct-injection engine equipped with a pilot injection and a standard diesel fuel pump-tine-nozzle system for injection of the CWS. Gurney (62) performed experiments in a single-cylinder research engine. Brehob and Sawyer (63) observed slurry combustion in an optically accessible research engine. Likos and Ryan (23) used both CWS and coal powder in a high-swirl, high temperature pre-chamber engine, as did Kakwani (64, 65). The vast majority of the engine experience, however, has been reported by Hsu et al (35, 66-71) and Rao et al., (28, 72) each using two different 4-stroke medium-speed engines. Schwalb (2l) reported successful operation in a 2-stroke high-speed diesel engine operating in compression ignition mode. Basically, only three different types of engine designs were considered in the METC program. The vast majority of the engine development work was done in the four-stroke designs using very open combustion chambers and very low swirl ratios. The General Electric and the Cooper/Arthur D. Little engine development work were both based on this type of design. The SwRI-General Motors and the SwRI-Detroit Diesel work were both performed using two-stroke designs that incorporated the use of port scavenging and four exhaust valves. The combustion chamber designs in these two-stroke engines were both very similar to the four-stroke engines, in that they used the open, "Mexican Hat" designs, with very shallow bowls. Some limited work was also done using divided combustion chamber designs at Adiabatics Inc. (15) and at SwRI (22).

As indicated above, the likelihood of fuel impingement on piston crown was very high in the open chamber designs used by GE and Cooper. In fact, Cooper found discoloration of spots on the piston crown at the center line of the fuel spray jets. However, fuel impingement was not associated with any problems, and in fact it may have been beneficial.

The goal of using a divided chamber design in the Adiabatics and SwRI work was to separate the injection and early combustion from the cold, lubricant wetted walls of the combustion chamber. These designs also offered the opportunity for utilization of thermally insulated surfaces to achieve higher temperatures and correspondingly higher ignition and combustion rates. Staged combustion for NOx control (via control of equivalence ratio) was also an objective. These designs were not pursued in the latter engine development activities primarily because they represented major changes in the head and piston designs of the category of engines being considered in the METC Program. These design changes would have required significant development work and would have also required very large investments to implement in a commercial engine. In addition, engine experience with CWS in the open chamber designs indicated that the combustion efficiencies were already very high, with ring and liner wear rates that could be controlled by appropriate selection of materials. Test results indicated that slurry fuel spray impingement on the piston crowns at GE and Cooper was not a problem.

The approach that was taken in all projects involved significant injection system development, followed by demonstration of operation on CWS in an engine, and then followed by system optimization in either single cylinder engines, or one cylinder of the actual development engine. The GE team and the Cooper/Arthur D. Little team both achieved several hundred hours of operation on CWS in medium speed diesel engines. The Detroit Diesel-SwRI Team achieved approximately 100 hours of operation in a high speed diesel engine. All of this work was performed using basically the same type of CWS fuel, consisting of a coal loading in the range of 50 percent by weight, with appropriate additives for control of sedimentation and viscosity.

The GE team based their system development on the GE Model 7FDL engine, which is a 1050 rpm engine with a bore of 229mm and a stroke of 267mm. Full scale engine development and testing were performed late in the program in a Dash-8 locomotive equipped with a 12 cylinder CWS engine. Cooper/Arthur D. Little performed the early (1985-91) system development work on a Cooper JS engine, which has a bore of 330mm, a stroke of 406mm, and an operating speed of 450 rpm. In 1992-93, system demonstration tests at Cooper were performed in a 6 cylinder LSC engine, which has a bore of 394mm, a stroke of 559mm. and a speed of 400 rpm. The high speed engine work at SwRI was performed in one cylinder of a Detroit Diesel 149 Series engine, which has a bore of 146mm, a stroke of 146mm, and a speed of 1900 rpm at rated power. The configurations of the two 4-stroke engines were very similar, as can be seen in Figure 62a and 62b, where cross-sectional schematics of both engines are compared. The 2-stroke, 149 Series engine used the high-speed work was port scavenged, with four exhaust valves, a central injection nozzle location, and a shallow combustion chamber. Each of the three manufacturers' engine test series with coal slurry fuels is described below.

General Electric

The initial engine experiments at GE were performed using a diaphragm separated fuel injection system that produced injection pressures in the range of 40 MPa at full load. Engine optimization included: variation in the intake air temperature from 85 to 110°C; variation of intake pressure in the range from 112 kPa to 329 kPa; and, simultaneous variation in the pilot and CWS injection timings. The early results indicated that combustion quality, based on coal burnout, was best at the highest possible intake pressure and temperature. They also found that the engine could be operated in compression ignition mode, but the performance was better if ignition was initiated using a pilot injection of diesel fuel, timed slightly in advance of the CWS injection (66). Coal burnout was reported to be in the range of 95 percent in these early experiments.

The combustion efficiency was increased to 99 percent and the conclusions regarding the injection timings were later modified after the compression ratio was increased to 13: 1 (68), and the CWS injection pressure was increased to 83 MPa (69) using an accumulator injection nozzle (Figure 58), and the number of holes in the injection nozzle tip reduced from 10 to 8 with the same total flow area (69). It was concluded that it was best to inject the CWS as early as 35° BTDC and to wait as long as possible to inject the pilot (120 BTDC) in order to allow time for mixing and evaporation of the water from the slurry. As the load is decreased, however, they found that the trend reversed, and it was better to inject the pilot before the CWS in order to provide sufficient heat for evaporation of the water and ignition of the coal. Hsu (71) verified by in-cylinder, high-speed photography that the high injection pressure resulted in impingement of the CWS on the hot piston crown, leading to improved atomization and more rapid combustion, if the pilot injection was delayed until 15° BTDC.

It appeared that pilot quantities equivalent to 5-6 percent of the total energy input at full load is required across the load range in order to achieve stable operation. Optimization of pilot orientation and configuration resulted in the system shown schematically in Figure 62b. The pilot nozzle utilized a tip configuration which the outer two holes had smaller diameter in order to maximize the interaction of the CWS jets with the pilot jets at the lowest possible pilot flow rate.

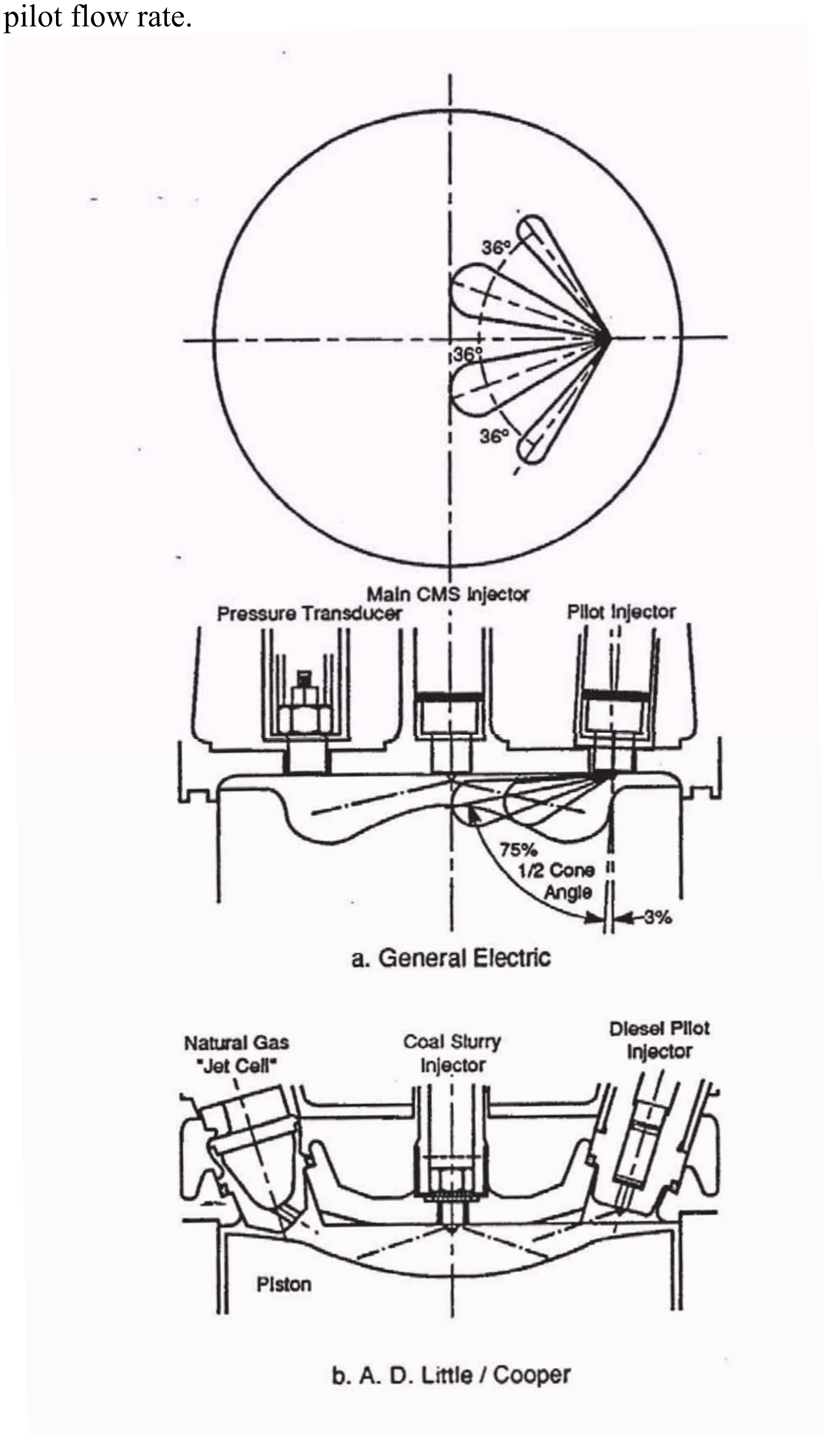


Figure 62. Engine Combustion Chamber Schematics for Operation on Coal-Water Slurry

The emission characteristics with CWS were generally the same in all GE engines, where the NOx emissions were reduced due to the charge cooling effect of the water. The particulate and the sulfur emissions were higher due to the presence of ash and sulfur, respectively, in the fuel. An emission control system was developed by to reduce the SO2 the NOx and the particulate emissions. The GE system included injection of copper oxide sorbent for the SOx and ammonia for catalytic reduction of NOx downstream of the turbocharger. A battier filter was installed at the end of the exhaust for removal of the particulate and the spent sorbent, for later regeneration (70, 87, 88).

Cooper/Arthur D. Little

System optimization in the Cooper/Arthur D. Little program was performed in a single cylinder version of the Cooper Model JS engine. The initial experiments were performed using the shuttle piston injection system designed by AMBAC (72). Using this system, they were able to achieve injection pressures in excess of 80MPa, even in the early work. Like the GE engine, the Cooper engines were generally turbocharged and intercooled. The coal engine configuration evolved to a system where the intercooler was bypassed so that the intake air temperature was in the range of 145 C (28) They found that increasing the intake temperature beyond this level was effective in improving operation in compression ignition mode. They also concluded, however, that pilot injection gave more reliable operation and that the 145 C intake air temperature was more than adequate when using a pilot to initiate combustion.

The optimum pilot characteristics in the Cooper 400 rpm engine are very similar to the GE engine. It appears that the best pilot quantity is the equivalent of 2-6 percent of the total energy input at full load. In this slower engine, they found that the best pilot timing is to inject at the same time as the CWS or at 18° BTDC. Duration of injection was approximately 30-40 degrees crank angle.

Injection system optimization work centered on development of reliable components and optimum configurations for combustion. The effect of hole geometry was examined at SwRI (73) in terms of the effects on atomization and mixing. Extensive engine experiments were also performed by Cooper/Arthur D. Little to determine the optimum configuration for durability of the nozzle and performance of the engine. These experiments included examination of the effects of the hole size, number, and orientation in the nozzle tip. The results showed that hole from 0.4 to 0.65mm were acceptable and the number of holes could range from 12 to 20 for this size engine.

The Cooper experience in the JS engine was successfully transferred to the larger bore LSC engine and demonstrated in a six-cylinder configuration. The LSB coal engine was turbocharged without an intercooler so that the intake air temperature was in the range of 135°C, with an injection pressure in the range of 100 MPa, and a CWS timing of 22° BTDC. The pilot quantity was in the range of 2-4 percent of the full power energy input, and was injected two points 180 degrees apart. The pilot timing was adjusted slightly in advance of the CWS injection.

The full LSC engine demonstration tests were performed using an extensive exhaust aftertreatment system that included ammonia injection and Selective Catalytic Reduction of NOx, sodium-based sorbent injection and collection for sulfur oxides control, and the use of a bag house for sorbent recovery and particulate removal.

Detroit Diesel/SwRI

Southwest Research Institute (25) performed CWS experiments early in the METC program in both a 71 Series Detroit Diesel high-speed engine and a General Motors Electro-Motive Division (EMD) 567 medium-speed engine. The Detroit Diesel engine is geometrically a one-half scale version of the EMD engine. This early work, while limited in duration, pointed out the temperature advantage of using a 2-stroke engine for operation on CWS. In addition, Kakwani et al. (15) were the first to point out that high speed engine can be operated on CWS if the in-cylinder temperatures are sufficiently high at the time of CWS injection. Detroit Diesel and SwRI, therefore, set out in 1990 to demonstrate that high speed diesel engines could be successfully operated on CWS (82).

An initial economic analysis indicated that the use of CWS in large mining equipment and in select marine applications offered some potential economic advantage over the use of diesel fuel. (89) The Detroit Diesel 149 Series engine, commonly used in these applications, was selected for use in the initial feasibility experiments. This engine is geometrically very similar to both the 71 Series and the EMD 567, except that it generally equipped with electronic unit injectors and incorporates, in its standard diesel fuel configuration, a ceramic thermal barrier coating on the piston crown to allow the higher piston temperatures that accompany high output. The engine is port scavenged through four exhaust valves. Scavenging is accomplished using an engine driven blower and turbochargers. The air flow path is through the turbochargers, an intercooler, the blower, and then into the air box that surrounds the intake ports on each cylinder. The engine is equipped with a bypass valve that is used at some operating conditions to bypass the blower and charge the air box directly through the turbochargers. Control of the bypass rate affects the air box pressure, and in turn the scavenging ratio, the residual exhaust gas left in the cylinders. and the temperature of the air and residual gas charge in the cylinder at the start of the compression stroke. Controlling the bypass ratio is a very effective way of increasing the compression temperature, and is the primary design feature of this type of engine that has allowed Detroit Diesel to operate the engine in compression ignition mode on methanol.

The CWS development experiments were performed using one cylinder of an 8-cylinder engine. The head was modified to accept an in-cylinder pressure transducer and the piston was changed to increase the compression ratio from the normal 15:1 to 19:1, in order to increase the compression temperature. This change was made based on modeling work (74) that indicated that the higher compression ratio was needed in order to achieve the ignition delay times and combustion rates required for operation of the engine at 1900 rpm. A shuttle piston injection system was also developed for operation at these higher speeds. It is important to note the injection system on the 400 rpm Cooper engine operates at 200 injections per minute. The injection system on the 1000 rpm GE engine runs at 500 injections per minute, while the injection system on the DDC Model 149 engine had to operate at 1900 injections per minute, or almost 10 times as fast as Cooper system and 4 times as fast as the GE system.

The modeling work (74) indicated that the compression ratio had to be increased and a pilot injection of diesel fuel was required in order to achieve acceptable combustion rates and coal burnout. The results of the experiments indicated that reliable auto-ignition of the CWS could be achieved at the high load conditions if the intake temperature was made as high as possible. This was accomplished in the experimental engine by closing the blower bypass (route all of the air through the blower) and by stopping the flow of coolant to the intercooler. The problem was one of balancing these conditions across the speed-load range of the engine, when the in-cylinder temperatures tend to decrease as the load decreases. Injection timing was found to be critical for operation in compression ignition mode, where the peak: cylinder pressure was excessive if the injection timing was advanced too far, and misfire occurred due to progressively lowering temperature as the injection timing was either retarded or advanced to far in the cycle.

The best combustion results were obtained with an injection timing of 20° BTDC and the highest achievable air box pressure and temperature, or 125°C and 192kPa, respectively. These conditions were limited by the constraints of the available hardware and were probably not optimum. Under these conditions, operation at part load was erratic due to reduced temperature, and the maximum load was limited by the fact that the peak pressures exceeded the design limits of the engine. It is probable that higher temperature and pressure would have improved the part load operation and reduced the ignition delay time and the high combustion rates at the higher loads. Coal burnout under these conditions reportedly still exceeds 99 percent at the maximum load condition.

COAL-WATER SLURRY PROPERTIES

Coal-Water-Slurries (CWS) are two phase mixtures that generally exhibit complex flow and combustion characteristics. Important characteristics of these slurries include flow at no, or low, shear for settling properties of the solids, flow at high shears normally encountered in diesel injection systems, atomization and evaporation properties of slurry sprays, and ignition and combustion properties at conditions normally achievable in engines. These properties of the slurry are dependent on the coal rank and source, the particle size distribution of the solids in the slurry, the mass loading of the coal, and the types and concentrations of additives that are used to improve stability, compatibility, and flow at high shear rate. This section is not meant to be an all inclusive review of slurry technology, but rather a brief summary of the findings of the METC Program related to the more important interactions of the slurry properties and performance in the engine hardware.

There were at least eleven suppliers of CWS in 1985. Otisca Industries, AMAX, and United Coal were the main sources for the CWS used in the early METC sponsored projects. Many of these companies were not in the slurry business by 1988, and Energy International, University of North Dakota Energy & Environmental Research Center, CQ Inc., and Otisca supplied most of the slurry for the work completed in the program (1989-93).

CWS preparation involves several steps, including mining, shipping, crushing, cleaning, grinding, fine cleaning, and slurry preparation, additive blending, mixing, storage, and shipping. All of these processes add cost to the resulting CWS. The Cooper/Arthur D. Little team developed an economic model that was used to evaluate the different factors in the CWS processing stream that affects the cost. While the cost of the raw coal and transportation account for most of the cost of the CWS, coal cleaning and additive costs are significant. Cleaning technologies that were examined in the METC program included heavy media separation, coarse flotation, fine flotation, oil agglomeration, and chemical cleaning. Several different approaches to each of these techniques are possible, but the teams generally examined only one example of the techniques that were available.

CWS property effects were examined as a part of the engine development activities at GE, Cooper/Arthur D. Little, Detroit Diesel/SwRI, and Adiabatics Inc. The initial goals were generally to improve the handling and combustion characteristics as a part of the engine optimization. GE (19) examined three different sources of bituminous and two different sources of sub-bituminous coal. Blue Gem Seam of bituminous coal, chemically cleaned by Otisca to less than 1 percent ash and sulfur, was tested in two different sizes and with two different additive packages. Additives are generally used to control low and high shear viscosity, such as xantham gum and surfactant, respectively, and dispersants to prevent agglomeration.

The results of the GE studies indicated that while smaller coal particles produced faster burn rates, selection of the appropriate additive can be used to increase the burn rate of larger particles, at least for the Otisca cleaned coal. Ammonium lignosulphonate produced slightly faster burning than ammonium condensed naphtalene sulphonate dispersant.

They also found that there was not much difference in the burnout and burn rates of the chemically cleaned bituminous coals. A physically-cleaned bituminous coal from a different source (Kentucky Splint Seam) appeared to have a lower burnout than chemical cleaned coals. However, this slurry also had a much larger particle size distribution (8.2 micron mass mean particle size versus 4.6 micron).

The sub-bituminous coals generally burned slower than bituminous coals in the GE engine. It should be noted, however, that the heating values of these fuels are lower and the injection rates were not adjusted in the GE tests to account for the lower rates of heat addition to the engine. There were also some differences noted between performance of the various sub-bituminous coals.

The results of the Cooper/Arthur D. Little fuel evaluations (75) are summarized in Table 25. It was generally noted that the lower speed 400 RPM Cooper engine was not very sensitive to the CWS properties, as long as the slurry did not exhibit flow problems that prevented operation in the engine.

Table 25. CWS Properties Tested

Effect	Coal Property	Range Tested 3/28/90)	Results
Combustion Performance	Volatile content Rank Heating Value Particle Size	27-41% Bituminous/subbit. 10-15 kBtu/lb (dry) 10-85 μm top size 3-12 μm mean size	All Okay
Emissions Control Cost	Sulfur Nitrogen	0.7-1.0 1.2-1.8%	<2% Okay TBD
Handling	Solids Content Viscosity	48-55% 200-400 cp	All Okay
Wear, CWS Cost	Ash Content Hard Mineral Content	0.5-3.8% --	<1.8 Okay TBD

As shown in Table 25, particle-size distributions that have means in the range from 3 to 12 microns, and top sizes up to 85 microns are acceptable. The choice between coal ranks is also open, except that the heating value does have an impact on the injection requirements and must be considered when designing the injection system. Viscosity in the range of 200-300 centipoise does not appear to be a problem, but GE (19) reported that increased viscosity results in increased pressure drops in the injection system, and lower injection pressures.

The rheological properties of the CWS fuels sedimentation tendency of the solids during storage, the flow characteristics during transport piping systems and pumps, and the injection and atomization in diesel engine injection equipment CWS fuels are generally non Newtonian, exhibiting nonlinear relationships between the shear stress and shear rate, so that the viscosity is variable and dependent on mass loading and the shear rate. It should be noted that the apparent viscosities of the CWS fuels are strongly affected by the mass loading of coal in the slurry. Most CWS fuels exhibit a very dramatic increase in the apparent viscosity as the mass loading approaches the 50 percent level, depending on particle size distribution.

There was a relationship between the apparent viscosity and the shear rate for slurries formulated at SwRI using physically cleaned coal supplied by AMAX. Results were obtained for three different distributions, where the "as received" coal had a mean size of 10.2 and top size of 28 microns. "Ground once" coal had a mean size of 7.4 and a top size of 18.5 microns, and "ground twice" coal had a mean of 5.5 microns and top size of 12.5 microns. The high viscosity at low shear rate is desirable because this minimizes sedimentation. The slurries all exhibited shear thinning in the range from 1 to 30 sec-1, and relatively constant viscosity in the range from 30 to 1000 sec-1. The low shear rheology is advantageous for storage (high viscosity at no shear) and transport through piping system (shear thinning at low shear rate). The CWS with the larger particles exhibited shear thickening tendencies beyond 1000 sec-1, but the CWS with the smaller particles had nearly constant viscosity beyond 10,000 sec-l. As will be shown in the next section, the high apparent viscosities of the slurries at high shear rate have significant effects on the injection and atomization characteristics relative to atomization observed with diesel fuel.

INJECTION CHARACTERISTICS

Initial efforts at injection system design were aimed at solving the reliability and durability characteristics of the systems. It appears that the shuttle piston design offers the most promise for providing the fuel pressures required for acceptable atomization. The conclusion of the GE engine development work was that an accumulator nozzle design was required in order to achieve acceptable performance in the 1000 rpm GE engine (500 injections/min). Detroit Diesel/SwRI and Cooper/Arthur D. Little found that high burnout could be achieved over a very broad range of engine speeds, from 400 to 1900 rpm (200 to 1990 injections/min) using high injection pressures in conventional injection nozzles. It is apparent that the conventional nozzle approach with high injection pressure is acceptable for the lower speed (400 RPM) Cooper engine. Here the engine bore size demands high penetration, the engine speed allows time for this penetration, and nozzle components are large enough to accommodate more holes. The accumulator concept may improve the performance of the high speed DDC engine (1900 RPM), but it is unclear if the accumulator pressure can be maintained at the higher engine speeds.

Ignition and combustion of any fuel, including CWS, in a diesel engine are highly dependent on the fuel injection characteristics of spray tip penetration rate, spray cone angle (both of these reflect the air entrainment rate), and the drop size distribution in the spray. Drop size distribution and the surrounding air temperature determine the evaporation and heating rates of the fuel. Holve (56) pointed out that fine grinding of coal is not beneficial unless the injection is capable of producing some drops that are smaller than the larger solid particles in the slurry. He found (76) that the particle distribution in CWS sprays were larger than the maximum size of the coal particles in the slurry. Morrison (77) also noted that small particle are not always achieved during injection and combustion by having small particles in the slurry. It is theorized that the coal particles agglomerate to form much larger particles, up to three times the size of the original coal particles. (76)

The engine experiments in the Cooper/Arthur D. Little project indicated that ignition and combustion characteristics were only slightly improved as the fuel injection pressure was increased in the 45-70 MPa range. The GE team observed similar results in their experiments. The higher injection pressures, however, produced correspondingly higher velocities in the injection nozzles and higher wear rates. It was theorized that the primary atomization could be improved by modifications to the design of the nozzle holes, and thus allow the use of lower injection pressure.

As shown in Figure 63 from Dodge et al (73), the mean drop size for the slurry sprays ranged from 25 to 20 microns for various mass loadings of 15 micron mean size coal particles, and from 45 to 30 micron drops *for* a 52 percent mass loading of 14 micron mean size coal. The maximum coal particle size was 85 microns in these experiments. Kihm et al. (78) reported mean spray drop sizes for CWS in the range from 30 to 100 microns for injection pressure in the range from 28 to 110MPa, but the interpretation of these measurements was complicated by the fact that there was evidence in the data (high intensity signal on the inner ring detector of the laser diffraction instrument) of window fouling. Dodge et al. (73) found that the hole geometry has very little effect on the performance in terms of the cone angle and the drop size distribution.

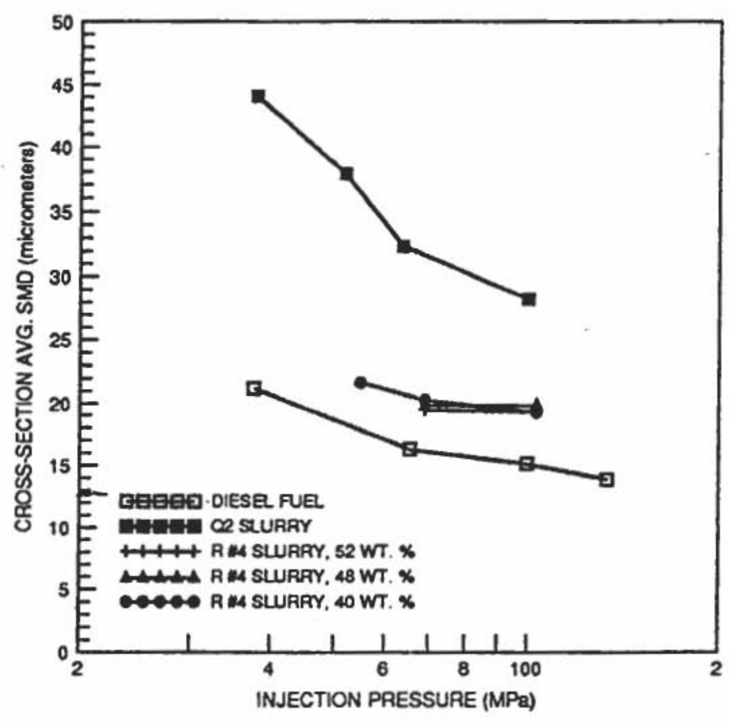


Figure 63. Average SMD of CWS Sprays Versus injection Pressure

ENGINE WEAR

Injection system wear problems were addressed above in another section. The other major wear problems in CWS engines occur in the ring and liner areas. The work of Mayville et al

(36), Ives (89), and Schwalb et al (79, 80) in reciprocating wear fixtures indicate that ring and liner wear is a combination of soft abrasive wear, due to the presence of coal in the wear couple, and three-body wear associated with the ash particles that enter the wear region. The three-body wear associated with the presence of the ash particles is a much more significant contributor to the wear rate than the soft abrasive wear, but it should be noted that ring and liner wear would still be a problem if unburned coal enters the wear couple, even if the ash were totally removed from the coal. Increased surface roughness was found to increases the thickness of the hydrodynamic oil film and detergent additives were found to minimize the abrasive wear, suggesting that surface finish and lubricant formulation could be tailored to minimize wear rates. (80)

The magnitude of the ring and liner wear problem demonstrated in Figure 64 from Mayville et al. (37) where the ring wear rates on CWS are compared to that required to achieve diesel fuel durability. Also shown on the figure are the wear rates for other material combinations, indicating that the wear rates are greatly reduced through the use of hard coatings.

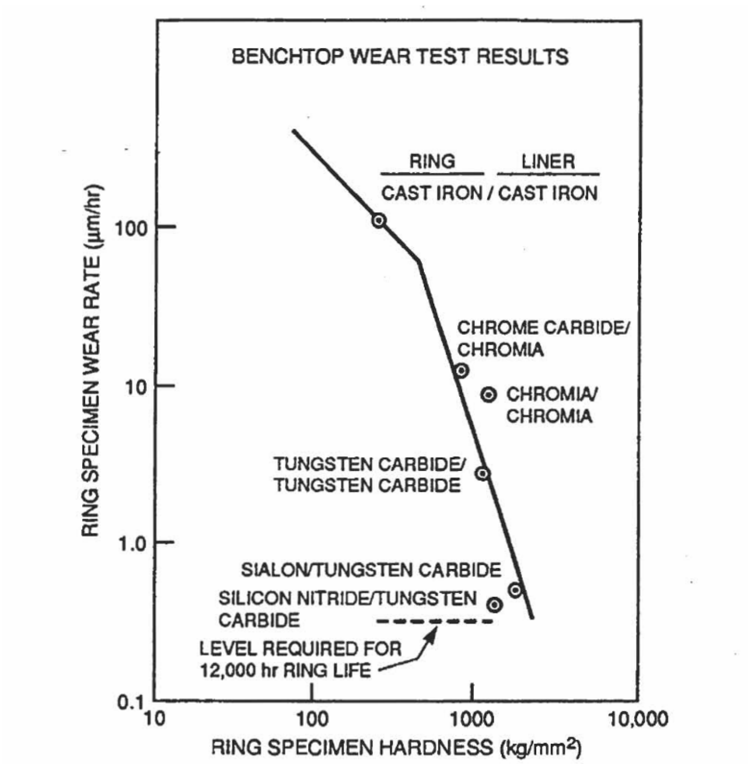


Figure 64. **Arthur D. Little Ring and Liner Wear Data**

The hard coatings were examined in some detail by both the Cooper/Arthur D. Little team and the GE team. The Cooper/Arthur D. Little work, performed by Mayville et al. (81), indicated that tungsten carbide coated ring and liners exhibited acceptable wear characteristics during operation on CWS. This conclusion is illustrated in the results shown in

Figure 65, where the end gap change is plotted versus run time for the coated and the standard components. Similar data from the team, reported by Flynn et al. (31), are presented in Figure 66 and Figure 67, for the liner and rings, respectively. The material combinations that were examined in the GE project (31) are summarized in Table 26. The GE team also concluded that the tungsten carbide coatings provided acceptable wear characteristics.

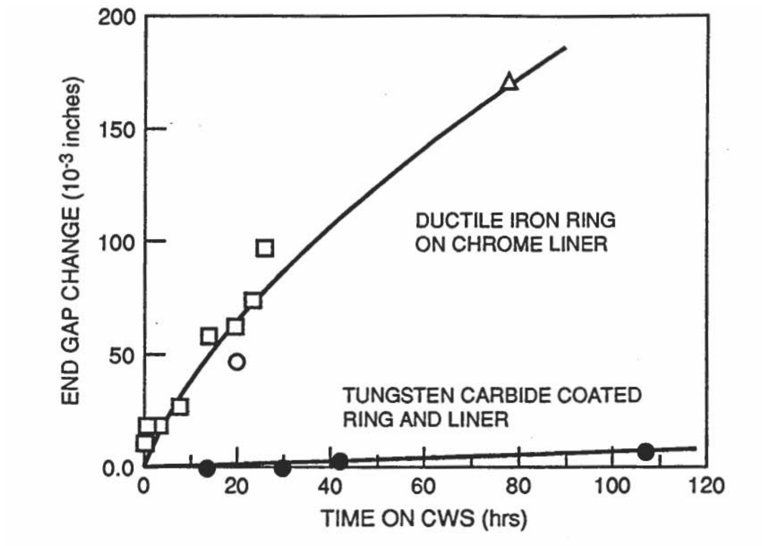


Figure 65. Ring and Liner Wear Rate Data for Conventional and Hard Materials

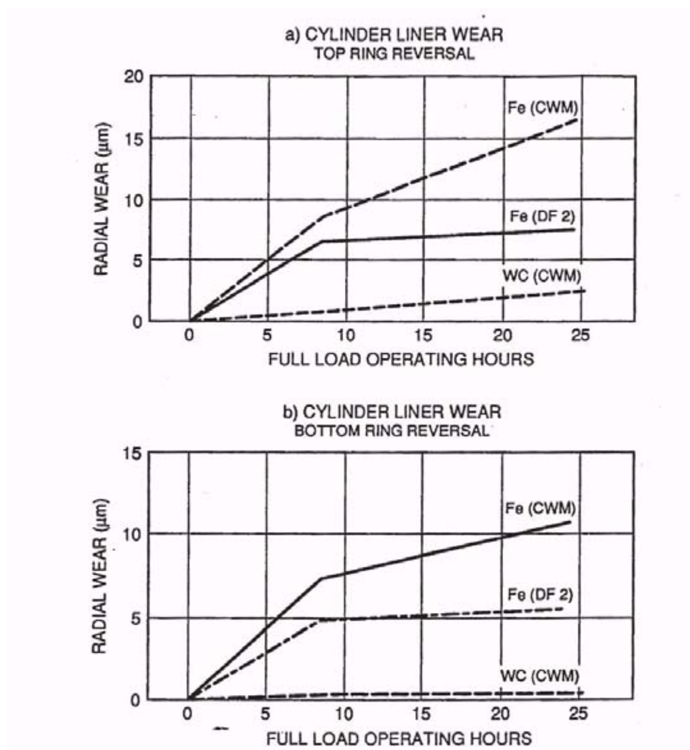


Figure 66. Liner Wear Rates for Various Materials

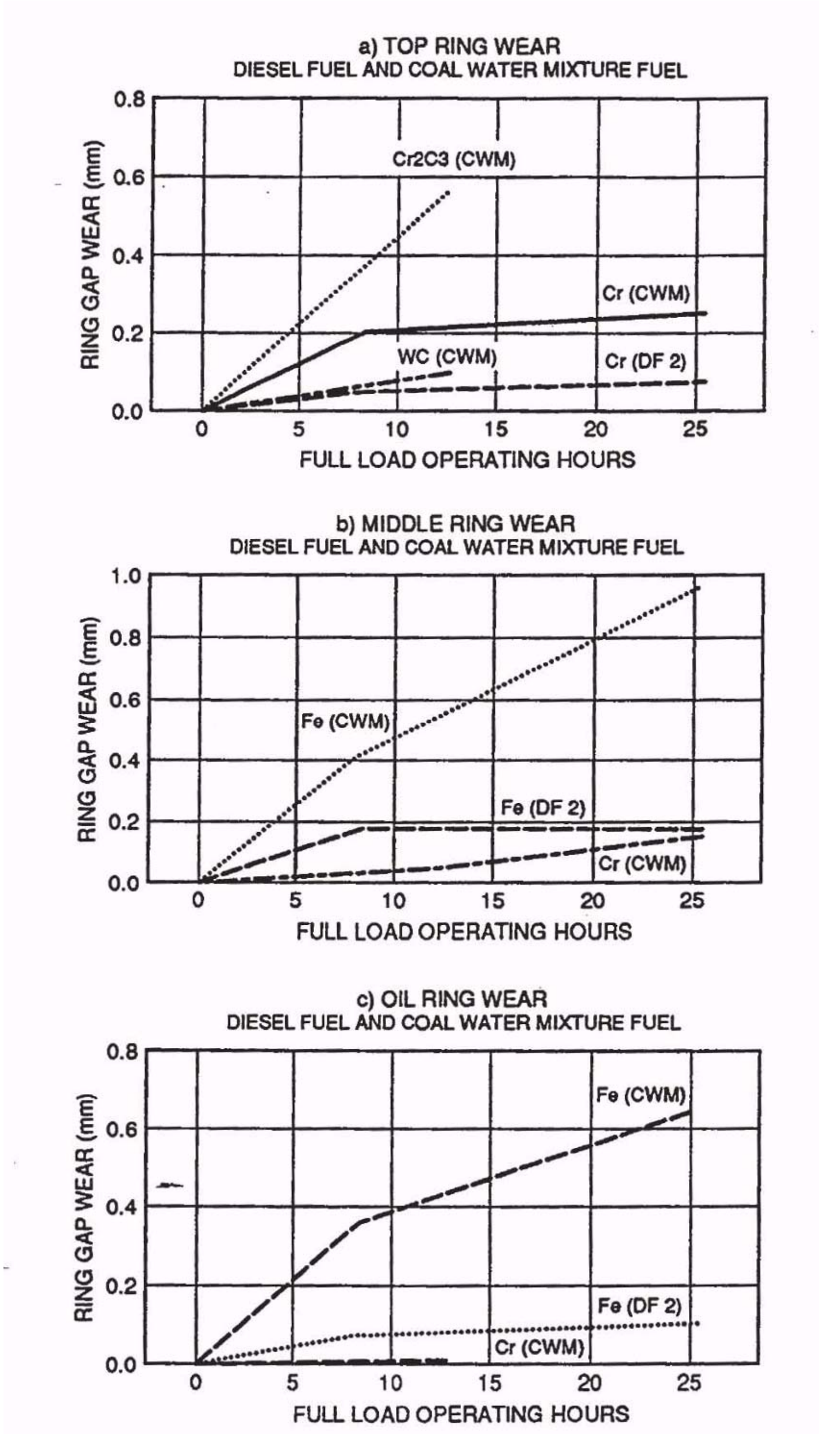


Figure 67. Ring Wear Rates for Various Materials

Table 26. Yanmar Engine Results

Run Number	Ring Material	Liner Material	Ring Wear Rate cc/hr
Uncontaminated Runs			
50	Chrome	Cast Iron	1.8E-5
51	WC+Co (ST)	WC+Co (LPPD)	1.4E-5
2%-D4 Contamination			
52	Chrome	Cast Iron	41.5E-5
53	WC+Co (ST)	WC+Co (LPPD)	55.0E-5
54	WC+Co (PT)	WC+Co (LPPD)	64.0E-5
56	WC+Co (K96)	WC+Co (K703)	0.8E-5
57	WC_Co (ST)	WC+Co (K703)	3.5E-5
58	Chrome	WC+Co (K703)	7.1E-5
59	Chrome	Cast Iron	28.8E-5
60	Cr3C2_Mo (K1008)	WC+Co (K703)	24.5E-5
Otisca Ash Contaminated			
61	WC+Co (K96)	WC+Co (K703)	1.43E-5
62	WC+Co (ST)	WC+Co (K703)	7.00E-5
63	Chrome	Cast Iron	250.00E-5
ST	Serma Tech Plasma Sprayed		
PT	Plasma Technics Plasma Sprayed		
LPPD	CRD Low Pressure Plasma Deposited		
K96	Kennametal Monolithic Carbide		
K703	Kennametal Monolithic Carbide		

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CWS ENGINE STATUS -1994

By 1994, the use of coal-water slurry in medium speed diesel engines had been demonstrated to be technically feasible at least for the limited run times attempted so far. This conclusion is based upon the use of well formulated coal-water slurries (CWS) in the range of 45-50 weight percent loading, using bituminous and sub-bituminous coals ground to 5-12 micron mean size particles, with a top size limit of 85 microns, and sulfur and ash contents range of 1 percent each by mass in the slurry (2% ash on dry coal basis). Higher ash contents may be usable in the engines, but the technology for coal cleaning has advanced to the point that 2 percent ash (dry coal basis) is reasonable for many bituminous coals. By 1994, several hundred hours of engine operation had been achieved in the GE program and over 1100 hours of testing was done in the Cooper/Arthur D. Little program. Long-term durability consistent with power generation plants (40,000 or more hours) is still yet to be demonstrated.

When the price differential between natural gas and raw coal increases above $2.50 to $3 per million Btu, the economic factors will appear favorable for use of CWS-diesels. That differential appears sufficient to cover the system incremental capital costs, emissions control, parts replacement, and operating costs. It appears that the proven level of exhaust aftertreatment based on test data is adequate to meet current and future emission standards.

The state of the technology in 1994 was such that the engine designs for coal slurry can be very well defined for engines that operate in the speed range below 1000 rpm (e.g., GE, Cooper-Bessemer, and Fairbanks Morse engines). For these engines, the next step is extended engine run times to demonstrate that hard component replacement rates are acceptable. The design for higher speed engines are not as well defined because the operating experience of these engines to date is less extensive. Below we outline the current understanding of CWS engines and by doing so, to clearly define the starting point when the fuel economics become more favorable.

Fuel Specification for Coal Slurry

The coal slurry fuel specification is somewhat dependent on the engine speed, in that the operating experience indicates that the slower 400 rpm engines are much less sensitive to factors than the 1000 rpm engine.

Fuel System Design for Coal Slurry

Details of the coal slurry fuel system design can be obtained above, or from the cited references. The following are the salient points that are common to all designs and should be considered as the starting point for future work in this area:

- Slurry Piping – smooth pipe, no dead volumes, no rapid in flow area

- Tank - Continuously recirculating, horizontal-cylindrical tank equipped with a floating suction, and discharge through a manifold with exit holes designed to produce flow with RexH=5.7

- Injection Pump - Convention diesel injection pump sized to inject the required amount of slurry to achieve full load and coupled to the engine, supplying diesel fuel pressure pulses to a shuttle piston assembly.

- Shuttle Piston Assembly - Convention injection pump barrel and plunger assembly design parameters of surface finish and clearances, titanium nitride coated, sized to displace 150 percent of the required full load slurry flow, and with an L/D of approximately 1.

- Nozzle Holes - Sapphire or diamond compact inserts.

- Needle Valve Seat - Tungsten Carbide seat insert.

- Needle Valve and Barrel - Tungsten carbide plasma coated steel

Engine Design for Coal Slurry Fuel

The results of the METC Program indicate that conventional open chamber, direct injection, low-swirl combustion chambers are acceptable for operation on CWS fuels. Combustion efficiencies of 99 percent or higher were routinely demonstrated in these designs, provided the intake temperature is raised to 135-180 C. The bowl shape can be either a shallow "Mexican Hat" design as used in the GE and Detroit Diesel-SwRI projects, or a shallow as used in the Cooper engine.

It is clear that turbocharging is required to achieve acceptable engine efficiency and breathing characteristics, especially with the higher intake air temperatures required for operation on CWS. The intake air temperature should be at least 135°C and the pressure should be in the range of 300kPa, while autoignition of coal was demonstrated in the GE, the Cooper, Adiabatics, and the Detroit Diesel engines, and repeatable ignition timing control with a pilot is essential for reliable and efficient operation of these a CWS engine. Pilot injection of small quantities of diesel fuel (2-5% of total fuel energy} appears to be the best method of ignition control. It offers the opportunity for cold starting and operation at the part load conditions, especially idle, where the demonstrated experience on CWS indicates that the engine temperatures drop below the levels required for reliable operation on CWS.

The rings and liner should be tungsten carbide coated and the lubricant should contain high concentrations of dispersant additive to prevent excessive wear of these components. It appears that filtration of the lubricating oil using the best available filter technology (pleated paper filter with 5 micron pore size) is sufficient to control wear rates the rest of the engine.

Appendix E. Coal Feedstocks for Engine Grade Coal-Water Slurry Fuels

1.0. INTRODUCTION

Currently, diesel cycle engines used for satisfying peak power requirements at electric power generating stations are fueled with petroleum. The reasons for this are simple. Fuel oil has been inexpensive, has a high energy-to-weight ratio, can be refined to a uniform specification, and is easy to introduce into the combustion chamber. As the cost of world petroleum resources increases and the United States moves to further exploit its own energy resources, alternative fuels will be sought for power generation needs.

Coal is another fuel which can be used in diesel engines. Like fuel oil, it is inexpensive and has a high energy-to-weight ratio. Each seam of coal, however, has an identity all its own. Refinement of any two or more coals to the same uniform fuel specification, short of a conversion process such as liquefaction or gasification, can be achieved only with great difficulty or not at all. Coal, in its dry pulverized form, has also been rejected as a diesel fuel due to its explosive nature and to the lack of a satisfactory method of delivery to the combustion chamber.

Coal slurries have been used successfully as a means of transporting coal in pipelines and show promise as a transport medium for coal in diesel applications. Slurry technology has advanced enough so that physical properties suitable for oil fired combustor retrofit applications have been met. In diesel engine applications, the high injection pressures required for atomization cause unacceptable nozzle wear when coal-water slurries are used. In the Arthur D. Little/METC Diesel Engine Development Program, component wear problems have been approached from the standpoints of hardware configuration, materials-of-construction, and fuel development.

It has been observed in this program that engine component wear can be directly related to mineral matter constituent concentration and that any reduction will positively affect component wear performance. A combination of nozzle component design, nozzle component material selection, coal selection, and coal beneficiation are being investigated to increase the lifetime of engine components and allow coal-water slurries to be effectively burned in a diesel engine.

The objective of the work performed in this study was to identify coals which can be beneficiated to acceptable mineral matter and sulfur concentrations utilizing currently known coal cleaning technologies. The cleaning steps required to produce an engine grade slurry from each promising candidate coal were estimated, as well as the current cost and availability.

2.0. INVESTIGATIVE METHODS

The overall plan for identifying coal feedstocks suitable for diesel engine applications was to:

1. Identify potential sources of United States Coal

2. Eliminate unsatisfactory candidates based upon a prioritized list of selection criteria.

3. Selected suitable candidates from different geographical locations for beneficiation

testing.

4. Identify availability and cost information for each candidate.

The portfolio of selected coals was intended to be representative of coal sources through-out the United States. It was not intended to be comprehensive.

2.1. IDENTIFICATION OF COAL SOURCES

The sources of information in this study include the Keystone Coal Industry Manual (1,2)* the PETC National Energy Technology Data Center, the Penn State Coal Sample Database, and data kept in AMAX R&D's files.

The Keystone Coal Industry Manual is a compendium of current coal operations in the United States. It lists over 5,000 mines in its Directory of Mines and over 2,000 minor companies in its Index of Smaller Operations. The manual alphabetically indexes coal mines by state and by company and lists information concerning coal seam mined, basic coal quality information, county in which the operation is located, and size of operation. Other sections give valuable information concerning raw coal quality (mainly as mined or washed ash and sulfur content, as well as proximate and Btu analyses) and quality trends in a state by state manner.

The PETC National Energy Technology Data Center database contains coal quality and washability data for hundreds of different coals.

The Penn State Coal Sample Bank contains samples and complete analyses for over 1,500 different samples of coal. The database can be searched over any combination of parameters (e.g., county and state, seam, volatile content, total and pyritic sulfur). For this study, the sample bank was used to procure specified samples of coal for fine coal washability testing.

AMAX has been performing coal beneficiation research and development since 1982 and has catalogued this information in AMAX Research & Development Center files. During this time, AMAX has identified a number of coal sources which provide product suitable for deep cleaning.

2.2. IDENTIFICATION OF CANDIDATE COALS

Since a given coal could meet all specifications for diesel combustion, except perhaps for one "fatal flaw" such as limited reserves, candidate coals were eliminated on a per case basis. The criteria for elimination of coals from the more than 5,000 mines listed in the Keystone Manual included:

1. Coals with a high organic sulfur content (greater than 1.0 weight percent).
2. Washability trends indicating sufficient ash reduction may not be attainable with fine coal beneficiation technologies.
3. Demonstrated limited coal reserves (less than 20 million tons).
4. Small mines (less than 100,000 tons per year)
5. Rank (low rank unacceptable).

* Reference numbers refer to list at the end of Appendix E (Page E-17)

6. Location (a transportation consideration).

Coals with high organic sulfur were eliminated since organic sulfur is not substantially removed using physical beneficiation technologies. This criterion was included because sulfur compliance is required in any new coal combustion technology.

The PETC database includes washability analysis for samples of coal crushed to various size fractions. The size fractions included are minus 1-1/2 inch by 0, minus 3/8 inch by 0, and minus 14 mesh by 0. These data were used to get an indication of the effect crushing has on ash liberation. A coal which shows significant ash liberation upon crushing or increased Btu recovery can be considered as a good candidate for fine coal beneficiation. If, for example, a 1-1/2 inch by 0 sample yields a 3.0 weight percent ash product at a specific gravity (s. g.) of 1.3 and when crushed to pass 14 mesh, yields a product with an ash content of 1.5 weight percent, then the coal was included as a candidate for further fine coal cleaning experimentation. Coals from this database which did not show washability improvement upon crushing were eliminated.

Coals which demonstrated favorable washing trends often originated from seams or mines with limited reserves. The availability of these coals were checked against coal reserves data obtained from the Keystone Manual and two DOE/PETC reports (3,4) Coals with limited availability were eliminated from consideration and alternative sources (such as larger mines in the same proximity mining the same seam) were sought.

Lignites and sub-bituminous coals were not considered viable candidates, since coal loadings acceptable for diesel applications are not presently attainable short of some type of thermal or hydrothermal processing.

2.3. COAL SAMPLE PROCUREMENT AND FINE COAL BENEFICIATION AMENABILITY TESTING

The volume of data concerning fine coal beneficiation is limited and, in general, somewhat specialized, as are the accompanying analysis. It was desired that as many current samples of promising candidates as possible be procured such that they could be tested and compared on a uniform basis. Samples of coal for fine coal beneficiation amenability testing were obtained from a variety of sources. These sources include mines operated by AMAX Coal Industries, other mining companies such as Consolidation Coal Company, coal samples at AMAX R&D, and the Penn State Coal Sample Bank.

Coal preparation for these samples consisted of dry milling to 100 percent passing 200 mesh (75 microns). The coals were then suspended in an organic true heavy liquid (CERTIGRAVtm) for float-sink testing. The separations were performed using a laboratory centrifuge at specific gravities of 1.3 and 1.9. The float products were collected and analyzed for ash and sulfur content.

True heavy liquid (THL) separations for fine coal can give somewhat misleading results, especially for lower rank coals or coals with higher porosity (3,4) In these cases, the organic THL can penetrate the pores and increase the apparent specific gravity of individual coal particles. This may reduce the float product recovery of a given coal.

A good example of this phenomena is Indiana No.7.

The results of ash and sulfur analyses from the THL separations testing were used to give an indication of the relative cleanability of coals in the short list of candidates.

Several samples of coal were selected for fine coal beneficiation testing using froth flotation. This was done both for testing the effectiveness of froth flotation as a cleaning method and for comparison to the THL measure of fine coal cleanability.

Samples of these coals were milled to minus 325 mesh and subjected to single stage froth flotation using either a laboratory flotation cell or a 20-foot tall column flotation cell. Preliminary results indicated that a single stage would not prove sufficient. Subsequently, further testing of multi-stage flotation was performed.

2.4. COAL AND TRANSPORTATION COSTS

Coal and transportation costs are often considered to be confidential information by coal and rail companies. Exact information for a specific coal and a specific destination generally must be obtained by initiating contract negotiations for actual delivery. For this study, coal and transportation costs were obtained by examining published contract prices (5). The prices quoted are in dollars-per-million-Btu.

3.0. RESULTS

Here we present information about coals mined in various states, concerning the county(s) in which the seam is mined, raw coal data, washability, coal and transportation costs, and reserves. (6)

3.1. OVERVIEW

It was originally hoped that candidate coals could be located within the same state as the prospective market for coal-fired diesel engine power generation. Such proximity of coal to market was envisioned to reduce transportation and ultimate coal slurry costs. However, it became clear that criteria other than feedstock location had the largest impacts on coal slurry costs. These criteria included sulfur content, washability, and size of mines. While the data on candidate coals are organized geographically in this survey, the pairing of coal and power markets will not necessarily be made based upon geographic location. The coal source for the markets are, in this report, regarded more in terms of Eastern and Western coals. Information on coal quality and transportation costs is presented. Below are the considerations for making this judgment:

1. The impact of coal quality on diesel engine performance, wear, and sulfur emissions compliance is of major concern.

2. Coal quality varies regionally. In general, every coal mined in the Southwestern Virginia Coal Field or Southern Coal Field of West Virginia will be lower in sulfur and have better washability than any coal in Pennsylvania.

3. Coal transportation costs between points in the Eastern states range between \$0.15 and \$0.60 per million Btu and are not necessarily proportionate to distance shipped. The total (coal and transportation) cost differential between shipping a Virginia and a Pennsylvania coal to Boston will be between \$0.10 and \$0.30 per million Btu. Many Northeastern power plants have found that shipment of these low sulfur coals for blending with higher

sulfur coals is an economical means of meeting sulfur emission targets.

The cost differential between cleaning a coal produced close to its market but of lesser quality would likely be greater than the cost of transporting a coal of higher quality. The current state of commercialization of fine coal cleaning processes, coupled with cost variations within the coal industry, make this difficult to determine quantitatively.

3.2. COMPARISON OF DATABASE SEARCH WITH FINE COAL BENEFICIATION TESTS

Much of the data found in the PETC coal database and the Penn State database is over 10 years old and as much as 25 years old. There was concern that data of this vintage might not be relevant to the coals produced today. It was found, in general and excepting some of the Northern Appalachian coals, that a seam mined in a given state and county has a relatively consistent analysis and washability overtime. Table 1 shows washability data for samples of coal taken from seams in the same county at different times.

Table 1. Washability of Coals from the Same Seam Sample at Different Times Washed at a Specific Gravity of 1.3

Year Sampled	Coal Seam	State	County	Btu Recover	Ash, Weight%	Sulfur, Weight%
1989	Lower Kittaning	OH	Harrison	86.7	3.1	0.82
1973	Lower Kittaning	OH	Harrison	86.7	2.7	0.89
1980	Pittsburgh	WV	Harrison	66.0	3.8	1.9
1967	Pittsburgh	WV	Harrison	67.1	3.7	1.6
1980	Upper Freeport	WV	Preston	60.4	3.7	0.73
1969	Upper Freeport	WV	Preston	55.0	3.2	0.75
1985	Splash Dam	VA	Russella	80.0	2.4b	0.59
1968	Splash Dam	VA	Dickensona	78.0	3.6c	0.79

a. Adjacent countries.
b. Coal crushed to pass 200 mesh
c. Coal crushed to pass 14 mesh.

The data suggest that older coal washability data can be useful in the search for coal feedstocks. Since a limited number of coals were actually tested in this effort and since it was not desired to exclude coals of similar quality from consideration due to the lack of current sample data, washability analysis from the PETC database was included in this report which was produced as much as 15 years ago.

3.3. FORMAT OF THE DATABASE

In the database coals are organized by state, and within each state, the data sheets are arranged from highest quality to lowest. For the most part, the coals contained in the database can be cleaned with varying types of physical processing (from heavy media washing to agglomeration) to concentrations of ash and sulfur acceptable for diesel engine combustion (less than 2.0 weight percent ash and 1.0 weight percent sulfur). Coal reserves listed reflect the known recoverable resources for the county(s) listed. In many cases, reserves of the given seam may be considerably larger if adjacent counties were to be included.

3.4. COAL QUALITY AND RESERVES IN THE STATES STUDIED

Low sulfur coals are found in states or areas of states which were not covered by inland seas or oceans during or immediately subsequent to coal formation. Thus, coals found in the Middle Atlantic and Western states tend toward lower sulfur content.

Alabama

This state has large reserves of low sulfur coals which respond moderately well to physical cleaning, especially agglomeration. The Blue Creek and Black Creek seams yield coals with a high Hardgrove Index (soft coal) and thus have low grinding costs and readily make very stable slurries. However, these coals are low in volatile content which may limit their usefulness in diesel engine applications.

Arizona

Arizona has large reserves of low sulfur, (very) high volatile bituminous coals. All mining is done by the Peabody Coal Company to provide coal for two power generating stations under long-term contracts. Not a great deal of fine cleaning work has been done on Arizona coals by AMAX. However, the sample of coal tested for this study responded moderately well to fine coal cleaning (i.e., ash reduction to 3.0 weight percent).

Colorado

Colorado holds huge recoverable reserves (greater than 17 billion demonstrated recoverable tons and over 400 billion tons total) of low sulfur, high volatile bituminous coals. Uinta Basin coals may prove to be the best for diesel engine applications based upon experience with Colowyo coal.

Illinois

The bulk of the vast reserves of Illinois coals are very high in sulfur. There are limited reserves of an Illinois NO.6 coal (commonly called Herrin) in Sesser County that can be cleaned to low ash and 1 weight percent sulfur.

Indiana

There are large reserves of both high and low sulfur coals in Indiana. Coals such as Indiana NO.7 (AMAX Minnehaha Mine) and Brazil Block clean to the 2.0 percent range for ash and less than 0.6 weight percent sulfur.

Kentucky

The state of Kentucky has two distinctly different coal fields separated by a non-coal producing portion of the state. The Western Coal Field belongs to the Interior Coal Province (the Illinois Basin). This vast resource contains about 38 billion tons of medium to high sulfur coal (1.5 to 5.0 weight percent).

The Eastern Coal Field covers about twice the area of the Western Field and contains reserves of roughly 55 billion tons. The region is divided into six coal producing districts: Princess District, Licking River, Big Sandy, Hazard, Southwestern, and Upper Cumberland. Of the six districts, the coals of interest for diesel engine applications may come from the Upper Cumberland, Big Sandy, and Hazard districts, in that order. Some seams of interest with large reserves are Hance, Harlan, Upper Elkhorn Nos. 1, 2, and 3, Lower Elkhorn, and Blue Gem.

The coals found in the Upper Cumberland District are of the same high quality (high volatile, low ash, low sulfur, and amenable to fine coal beneficiation) as those found in the Southwestern Virginia Coal Fields and the Southern Coal Field of West Virginia. That is, these three coal fields are members of the same coal field which has been arbitrarily divided by state lines.

Ohio

Most Ohio coals are classified as either medium or high sulfur coals (1.5 to 6.0 weight percent). The relatively small reserves of low sulfur coal such as Lower Freeport (Ohio 6A) have been depleted. The highest quality coal with any significant reserves is Lower Kittaning in Mahoning or Holmes County; ash reduction suitable for diesel applications using physical cleaning methods is possible with sulfur in the 0.5 to 2.0 weight percent range.

Pennsylvania

A coal quality trend in Pennsylvania is that sulfur content increases northward and westward. Many of the familiar Pennsylvania seams (Brookeville, Upper, Middle, and Lower Kittaning, Upper and Lower Freeport, and Pittsburgh) are also mined in Ohio and West Virginia. These seams are extensive in the area they cover and have significantly different quality depending upon where it is mined. Parts of the Middle Kittaning seam wash very easily to about 1.0 to 1.5 weight percent ash and 0.5 to 0.7 weight percent sulfur, while other locations can achieve no better than 4.0 percent ash and 2.0 percent sulfur (using the same cleaning technique). Essentially the same thing can be said of any of the other seams mentioned above. It is quite possible for Pennsylvania coals to be used for diesel engine applications, but care should be taken in choosing the mine the coal will be purchased from.

Tennessee

The state of Tennessee apparently has good coals similar to eastern Kentucky and southern West Virginia; however, no Tennessee coals were tested in this study.

Utah

The state of Utah holds enormous coal reserves. There are roughly 10 billion tons in the Wasatch Plateau (Hiawatha, Castlegate, and Sunnyside coals) alone. These coals are low in sulfur, high in volatile content, and respond reasonably well to fine coal cleaning. Ash reduction to 1.35 weight percent has been shown in Rock Canyon coal and 1.99 weight percent in Hiawatha at sulfur contents of less than 0.4 weight percent.

Virginia

Coals mined in the Southwestern Virginia Coal Field are generally very high quality, low sulfur coals. Washability data show that seams in the Wise Formation such as Taggart, Dorchester, and Kelly can be cleaned to ash and sulfur levels suitable for diesel engine applications, often with only standard heavy media washing. The coal quality in this formation is similar to that found in the Upper Cumberland District of Kentucky and the Pottsville Group in the Southern Coal Field of West Virginia; the three coal producing districts are adjacent to each other and have arbitrarily been divided by state lines.

West Virginia

There are two distinct coal regions in West Virginia: a Northern Coal Field containing thin seams of generally medium to high sulfur (1.5 to over 3.0 weight percent) and medium to high ash (higher than 6.0 weight percent) and a Southern Coal Field containing thick seams of coal uniformly low in sulfur (less than 1.5 weight percent) and ash (less than 6.0 weight percent). The two coal fields are separated by a distinct "hinge line" which runs northeast to southwest through the middle of the state.

West Virginia ranks third in coal production behind Wyoming and Kentucky. Vast reserves of coals in the Pottsville Group and Pocahontas Formation are similar to coals found in the Upper Cumberland District of Kentucky and the Wise Formation of Virginia. Pocahontas NO.3 is a good example of a coal with large minable reserves (2.8 billion tons) and low as-received ash (average 4.0 percent) and shows good fine coal washability (ash reduction to 1.5 weight percent).

Wyoming

The state of Wyoming is estimated to contain as much as one-quarter of all United States coal reserves ranging in rank from lignite to high volatile A bituminous. The coals of interest for slurry production (bituminous) occur mainly in the Hanna and Bighorn coal basins, although mining in the Bighorn Basin amounts to just 0.1 percent of total production. No fine coal cleaning work was performed on Wyoming bituminous coals; however, for initial coal beneficiation work, a good starting point might be Bed 65 seam coal (see Section 3.7, "Suggested Coal Feedstocks").

3.5. PHYSICAL CLEANING TEST RESULTS

In this report, most of the data on a given coal's response to fine cleaning was obtained using centrifugal separations of fine coal suspended in an organic true heavy-liquid (THL). This method was used to determine the ash reduction possible through ash liberation via fine grinding. The information gained from these experiments was not meant to be used in place of experimental data for any other fine coal cleaning method. One of these fine coal cleaning technologies proposed for producing an ultra-clean engine grade fuel was flotation. To test the method in the context of this study, a series of flotation tests were performed on three promising Eastern bituminous feedstock candidates, a Western bituminous, and a Western sub-bituminous coal.

3.5.1. Preliminary Tests with High Splint Coal

The first set of experiments was performed on Kentucky High Splint coal (a baseline coal for all A. D. Little/METC diesel engine development efforts thus far). Since the floatability of coals vary widely depending upon rank (9) the experiments were performed to determine the proper amount of methyl isobutyl carbinol frother (MIBC), collector (kerosene), and particle size (75,50, or 25-micron top size) to achieve optimum ash reduction performance. The results of these tests are summarized in Table 2. Ash reduction performance utilizing heavy media washing is included for comparison.

All tests were performed using a 5-liter mechanical laboratory flotation cell. Each slurry for flotation cell testing was prepared by suspending 375 grams of raw coal in 4 liters of water. The product ash content is the average of all samples taken during the first 6 minutes of flotation. It was found that flotation provided ash reduction from 16.2 weight percent in the raw coal to about 8.0 weight percent in the Test Number 3 product. In comparison, heavy media washing gave a product with an ash content of about 2.5 weight percent. A test was subsequently performed using a 20-foot flotation column using parameters corresponding to Test No.3. The test produced a float product containing 5.9 weight percent ash.

Table 2. Results of Experiments to Determine Optimum Flotation Conditions for Kentucky High Splint Coal

Test	Grind Microns	MIBC ml	Kerosene, ml	Ash, Weigh %	Btu Recovery, %
Raw Coal				16.2	
No.1	25	1.35	0.35	8.5	75
No.2	75	1.35	0.35	9.7	46
No.3	25	1.65	0.35	8.0	69
No.4	75	1.65	0.35	9.2	65
No.5	25	1.35	0.80	8.1	71
No.6	75	1.35	0.80	9.3	88
No.7	25	1.65	0.80	8.4	59
No.8	75	1.65	0.80	8.7	85
No.9	50	1.50	0.50	9.1	23
No. 10	50	1.50	0.50	9.0	53
Heavy Media*	*-1/2* Inch x 0			2.5	60 - 70

* Heavy media washing at an apparent specific gravity of 1.3.

3.5.2. Flotation of Upper Elkhorn NO.3

A second set of experiments was carried out using Upper Elkhorn No.3 (UE3) coal. This coal was chosen in light of past experience with this seam and its relative ease of washing. The objective of these experiments was to test the effect of coal seam on flotation performance. Using heavy media washing, this coal yields a product with an ash content of approximately 1.25 weight percent. The results of the flotation tests are summarized in Table 3.

The best result was obtained in Test No.2. As with High Splint, flotation cell tests led to ash reduction of a little better than 50 percent. In this case, flotation achieved ash reduction from 7.3 weight percent in the raw coal to about 3.0 percent in the product. Heavy media washing on this coal reduced the ash level to approximately 1.25 weight percent.

3.5.3. Staged Flotation with Upper Elkhorn Coal

It was suspected that the poor performance of froth flotation as a method for cleaning coal was due in part to entrainment of ash in the froth. A second set of tests were performed on Upper Elkhorn No.3 coal to determine the extent of this effect.

Table 3. Upper Elkhorn Coal Flotation Test Matrix

Test	Grind	MIBC, ml	Kerosene, ml	Ash, Weight %	Recovery, Weight %
Raw Coal	--	--	--	7.3	--
Heavy Media Washing	-1/2 Inch	--	--	1.2	60 – 70*
No.1	Medium	1.50	0.50	3.1	94
No.2	Coarse	1.35	0.35	2.9	93
No.3	Fine	1.35	0.35	3.4	85
No.4	Fine	1.35	0.50	3.6	95
No.5	Fine	1.65	0.80	3.6	96
No.6	Coarse	1.50	0.50	3.0	95
*Estimated.					

The method used to test for ash entrainment with the froth was to add a second stage of flotation. For the tests, samples of the flotation product from the six single-stage experiments were combined to make a feed slurry for a second-stage test. The optimum reagent levels as determined from the single-stage tests were used. The results are summarized in Table 4.

Table 4. Two-Stage Froth Flotation with Upper Elkhorn No.3 Coal

	Weight% % Ash (Dry Basis)	Weight % Recovery	Cumulative % Btu Recovery
Raw Coal	7.3		
Stage 1 Product	3.7	93	97
Stage 2 Product	2.0	92	91

3.5.4. Staged Flotation with Kentucky High Splint

Further verification of this effect was sought by performing a three-stage test using Kentucky High Splint coal. Each stage was divided into five product cuts. The flotation product was collected over the following intervals: Product gathered from 0 to 1.5 minutes. Product gathered from 1.5 to 3.0 minutes. Product gathered from 3.0 to 6.0 minutes. Product gathered from 6.0 to 12.0 minutes. Product left in flotation cell after 12 minutes.

For each stage, the product gathered from 0 to 6 minutes was combined to make feed for the next stage. The data gathered from this work are listed in Table 5.

Table 5. Three-Stage Flotation Raw Data, Kentucky High Splint Coal

Collection Time, Minutes	Stage 1		Stage 2		Stage 3	
	Recovery, Weight %	Ash, Weight %	Recovery, Weight %	Ash, Weight %	Recovery, Weight %	Ash, Weight %
Feed Slurry	--	20.4	--	12.7	--	8.7
0 – 3.0	81.4	10.2	86.3	7.2	90.5	4.8
3.0 – 6.0	8.3	38.8	8.0	24.5	6.4	18.4
6.0 – 12.-	4.3	81.6	2.8	68.4	1.8	58.7
+12.0 (tails)	6.0	91.9	2.9	90.3	1.3	88.5

A material balance was made for this test to summarize the performance of the three stages under these operating conditions. The results are listed in Table 6.

Table 6. Three-Stage Froth Flotation with High Splint Coal

	Weight% % Ash (Dry Basis)	Cumulative Weight % Recovery	Cumulative % Btu Recovery
Stage 1 Feed	20.4		
Stage 2 Feed	12.7	89.7	98.4
Stage 3 Feed	8.7	84.6	97.1
Stage 3 Product	4.8	83.8	90.2

By comparing Stage 2 feed (Table 3) with Stage 2 product (Table 3), it can be seen that ash is reduced between stages by about one-third to one-half. The same is true for Stages 1 and 3. The results of these tests show that ash entrainment with the froth is significant.

Flotation with High-Shear Mixing Between Stages

Another concern with single and multi-stage flotation is possible conditioning of the ash with the collector. Since the coal and froth were vacuum filtered between stages, coal, ash, and kerosene were brought into close contact as a filter cake before being repulped in the stage of flotation. It was suspected that mineral matter associated with the product might stick to coal particles or even possibly coated with the collector. If this were the case, repulping the product in a high-shear environment would be necessary to "reliberate" the mineral matter.

To detect this effect, two tests were performed on the High Splint Stage 3 product. The first, designated "Stage 4", consisted of repulping the Stage 3 product coal and performing a lab flotation test. Samples of the product were taken for analysis. The balance of the Stage 4 product was repulped and mixed in a laboratory stirred-ball mill for approximately a minute and a half using 5-millimeter stainless steel balls. The slurry was then subjected to a fifth stage of flotation. Table 7 summarizes the analysis of the Stages 3, 4, and 5 product collected in the first minute and a half of flotation.

Table 7. High Splint Multi-Stage Flotation. % Ash in Float Product Gathered in the First 1.5 Minutes of Flotation

Test	Weight % Ash
Stage 3	4.3
Stage 4	3.8
Stage 5	2.8

The results show that milling between stages appears to improve ash reduction.

3.5.6. Staged Flotation with Kentucky Blue Gem Coal

A three-stage laboratory flotation test was performed on Kentucky Blue Gem coal. The objective of this test was to produce an engine grade coal product using flotation as a cleaning method. The task was not expected to be difficult, since the analysis of this raw coal ranges from 1.2 to 2.1 weight percent ash. The results of the three-stage flotation are listed in Table 8.

As expected, ash reduction to less than 1 weight percent provided little difficulty while maintaining high Btu recovery.

Table 8. Multi-Stage Flotation with Blue Gem Coal

Stage	Feed Ash, Weight %	Product Ash, Weight %	Cumulative % Btu Recovery
Stage 1	1.4	1.0	97.0
Stage 2	1.0	0.8	95.0
Stage 3	0.8	0.7	87.0

3.5.7. Western Coal Flotation Tests

Experiments were performed to test the floatability of two selected Western coals. One was a sub-bituminous coal (Wyodak) and one was a high-volatile C bituminous coal (Colowyo). The results of these tests are summarized in Table 9. Again, each test slurry was prepared by suspending 375 grams of raw coal in 4 liters of water.

Table 9. Western Coal Flotation Test Matrix

Test	Grind	MIBC, --ID	Kerosene, ml	Ash, Weight %	Recovery Weight %
Wyodak Feed				7.1	
Wyodak Float	Fine	3.5	1.80	5.8	
Wyodak Float	Fine	5.0	3.60	5.7	3
Wyodak Float	Fine	5.0	5.00	*	
Colowyo Feed				3.8	

Test	Grind	MIBC, --ID	Kerosene, ml	Ash, Weight %	Recovery Weight %
Colowyo Float	Fine	2.0	0.80	2.9	
Colowyo Float	Fine	5.0	2.20	2.6	6
Colowyo Float	Fine	7.0	3.00	2.7	8
Colowyo Float	Coarse	5.0	2.20	2.6	8

*Could not filter

The relation between rank and flotation performance shows itself here. The sub-bituminous coals showed very little cleaning and required excessive reagent quantities to achieve only marginal recovery. Colowyo coal required intermediate levels of frother and collector for flotation. Table 10 compares the proximate analysis of the coals tested in this study by column flotation.

Table 10. Column Flotation Test Coals Proximate Analysis

Coal	Moisture	Weight % Volatile (DMMF)*	Fixed Carbo (DMMF)*	Nominal Collecto Requirement (ml/100 grams)
Upper Elkhorn No.3	3	34.7	65.3	0.0
Blue Gem	1	39.7	60.3	0.0
High Splint	2	39.6	60.4	0.0
Colowyo	15	42.5	57.4	0.5
Wyodak	30	54.0	46.0	>1.3

*Dry, mineral matter free basis.

All three Eastern bituminous coals used 0.09 gram of kerosene per 100 grams of coal, but floatability decreased rapidly with increasing volatile content.

The high levels of kerosene and MIBC used in these Western coal flotation tests created handling problems (difficult to filter and dewater) and would be considered too costly on a commercial basis. Any future work on flotation of Western coals might begin with screening tests of effective frothers and collectors.

3.6. INTERPRETATION OF WASHABILITY DATA AND COAL CLEANING TESTS

Coal is a heterogeneous product. The sources are so numerous that the range of coal compositions available represents almost a continuum. Some coals clean to low ash with minimal cost, while others do not. There are coals which come out of the ground with nearly 50 percent mineral matter (the point at which many no longer consider it coal), which will clean to 2 percent ash using only heavy media washing (Winifrede in West Virginia, for example). There are others which are mined with 6 percent mineral matter and clean to no better than 4 percent ash. For these, it is necessary to apply advanced cleaning technologies, mainly either flotation or agglomeration.

These advanced cleaning technologies rely on mineral matter and pyrite separation from coal macerals by fine milling. The potential for ash and sulfur reduction via this means is highly specific to the coal in question. As a result, the determination of this potential tends to be highly empirical. Although actual mineral matter and sulfur reduction performance of these technologies can only be ultimately determined using the actual processing conditions, reduction performance resulting from liberation is hinted at in fine coal washability studies.

The particular coal feedstock selected for diesel applications is constrained by the fuel specifications which have been dictated by the engine, the most important specifications being mineral matter type and sulfur content. The ideal candidate coal for diesel engine grade coal-water slurry production would:

- Clean to less than 1.25 weight percent ash and less than 1 pound of sulfur dioxide per million Btu with minimal expense (e.g., utilizing standard coal preparation technology, jigs, heavy media cyclones, etc.).

- Be mined in the proximity of its power market.

- Have abundant reserves (e.g., a 20-year supply).

- Have a volatile content greater than 30 weight percent (to improve combustor performance).

Coals which satisfy these criteria do exist, but they represent only a small portion of total United States reserves. Expansion of the list of possible feedstocks depends upon a complex interrelationship of costs associated with preparing an “on-specification" coal slurry. Factors affecting the cost of diesel grade coal slurry include:

- FOB mine cost of coal

- Transportation costs

- Required processing steps

- BTU recovery

- Standard coal washing

- Milling (i.e., ash liberation cost for fine coal cleaning)

- Costs associated with flotation, agglomeration, etc.

This type of information was gathered for the coals investigated in this study, as an aid to help the reader select coals for their particular application. Interpolation or extrapolation of the data is discouraged. The reason for this is best illustrated by the following example. One coal may wash to 3 weight percent ash at 1/2 inch x 0 and to 1 weight percent ash at a size fraction of 14 mesh x 0. It does not follow that the coal will clean to 2 weight percent ash at a size fraction of 1/4 inch x 0 (i.e., the ash reduction performance may be no better than with the 1/2 inch x 0 size fraction). For this reason, it is suggested that the following procedure be utilized for selecting coals for diesel applications.

If standard coal preparation techniques (jigs or heavy media cyclones) are used:

- Select a coal such as Taggart from the sources suggested in Section 3.7.

If an advanced coal beneficiation technology is used:

- Select a different suitable coal from the desired state which shows fine coal washability, ash reduction to the 3.0 weight percent and sulfur reduction to the 0.8 weight percent ranges.

Procure a sample from the mine listed. If the mine is no longer operating, choose an alternative mine, preferably located in the same county.

Perform cleanability tests using the chosen advanced coal cleaning technology. If the coal does not perform satisfactorily, choose an alternate mine or seam.

3.7. SUGGESTED COAL FEEDSTOCKS

High quality coals suitable for diesel engine fuel preparation using physical cleaning methods can be found in just about every coal producing state. For commercial diesel engine power plants, the key issue is securing a stable long-term supply of "on specification" fuel. For such a plant, the findings of this report suggest a course of action with regard to coal selection:

In the East, large reserves of coals with very low sulfur content and excellent washability characteristics can be found in the Kentucky, Virginia and West Virginia tri-state region. The coal quality within a given seam of a given mine is very consistent when compared to other regions. The chances of obtaining a high quality coal from this area are high.

From this region, the following counties are likely to yield a suitable feedstock:

Kentucky	Floyd, Pike, Letcher, Harlan, Bell, Knox, and Whitley
Virginia	Lee, Wise, Dickenson, Buchanan, Scott, and Russell
West Virginia	Mingo, Logan, Boone, McDowell, Wyoming, Mercer, and Raleigh

From each state, the following coals are good first (but by no means the only) choices:

Kentucky	Taggart, Upper Elkhorn Nos. 2 and 3, and Blue Gem
Virginia	Taggart, Blair, Clintwood, and Dorchester
West Virginia	Lower War Eagle (also known as No. 2 Gas), Campbell Creek (also known as No.2 Gas, Peerless, and Upper War Eagle), Winifrede, Cedar Grove, Powellton, Kennedy, and Widow Kennedy

There are many potential sources of these coals and a few of the largest producers of these coals are listed below:

Kentucky	Scotia, Upper Taggart Mines Blue Diamond Coal Company

Virginia	Bullitt, Wentz, or Holton Complex Mines Westmoreland Coal Company
West Virginia	Sundial, Montcoal Mines Peabody Coal Corporation

These companies all mine several of the seams listed above.

In other geographic regions, the following coals and mines are suggested starting points:

Colorado	Recommended Seams: Colowyo Mine Colowyo Coal Company
Indiana	Recommended Seams: Indiana No. VII Minnehaha Mine AMAX Coal Company
Ohio	Recommended Seams: Lower Kittaning, Lower Freeport (6A) "K" Mine East Fairfield Coal Company
Pennsylvania	Recommended Seams: Brookeville, Lower Kittaning Mine Nos. 72-16 and 72-19 Perry Brothers Coal Company
Utah	Recommended Seams: Rock Canyon, Sunnyside Sage Point - Dugout Canyon Project Sunedco Coal Company
Wyoming	Recommended Seams: 60 Series Beds (especially Bed 65) Medicine Bow, Seminole II Mines Arch Mineral Group

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Source Key:
PETC
Penn State
AMAX

QUALITY AND WASHABILITY DATA FOR COALS INVESTIGATED IN THIS STUDY

Coal quality and/or washability data obtained from the Pittsburgh Energy Technology Center.

Coal sample and quality data obtained from the Pennsylvania State University Coal Sample Bank and Database.

Coal samples, quality, and/or washability data obtained through testing performed at the AMAX Research & Development Center.

Appendix F. Project Chronology

The Chronology of the Coal-Diesel Demonstration Project is shown in the table below.

Pre-Award Chronology	
1983 – 1992	• Team of ADL and Cooper developed pre-commercial prototypes of the coal-fired diesel technology with DOE support. Technical feasibility of full scale six-cylinder 1,800 kW engine proven in 100 hour test. Significant market potential seen in many parts of world for 10-100 MW coal-based distributed power systems.
1993	• The Clean Coal Diesel project was selected in Round 5 of CCT program. Cooperative Agreement with $38.3 million budget signed July 1994, negotiations led by Rita Bajura.
Easton MD Chronology	
July 1994 – March 1995	• Project team worked hard to make the coal diesel project happen at Easton MD host site. Completed EIV and initiated system design. Easton Utilities withdrew when Easton's load growth declined and they found they could purchase power at relatively low rates, rather than add capacity.
Fairbanks, Alaska Chronology	
Sept.1995 -1996	• DOE approved to resite in Alaska. Budget increased to $47.6 million. Neat tie into LRCWF project already under development at U. of Alaska. Added U. of Alaska and Usibelli Coal to the team. Modified Repayment Agreement.
July-Sept. 1997	• Performed new EIV for Alaska site and obtained FONSI.
May-Oct. 1997	• Fairbanks-Morse replaced Cooper-Bessemer as the coal diesel engine manufacturer. Modified Repayment Agreement.
June 1998	• Construction initiated at UAF on diesel/generator portion of project.
December 1998 – March 1999	• Manufacturing of Coal-diesel engine completed. Engine performance qualification tests were completed at FME factory. Coal Diesel engine delivered to UAF and installed in new diesel/generator building.
April 2000	• Start-up of the demonstration plant's Engine, Generator and Heat Recovery Steam Generator using diesel fuel
September 2000	• Acceptance testing for the demonstration plant's Diesel Engine Generator, Heat Recovery Steam Generator and the Selective Catalytic Reduction Unit was successfully completed.
December 2000	• Cost estimate to build LRCWF process plant in Alaska found to exceed available budget; Cooperative Agreement modified to include one-time purchase of LRCWF from a supplier rather than building and operating a plant (M011).
April 2001	• Manufacturing of hardened components completed for the two-cylinder Fairbanks Morse engine to allow LRCWF burning.
June 2001	• LRCWF injector tests were successfully completed at Fairbanks Morse.
November 2001	• The two-cylinder engine at Fairbanks Morse is ready for LRCWF performance tests.
Sept. 2002	• Search for Alaska CWF supplier is unsuccessful. Descoping discussions begun with DOE.
March 2003	• Preliminary framework for relocating the demonstration tests to be conducted on the Fairbanks Morse two-cylinder engine was presented at Project Review with DOE at Morgantown.
July 2003	• Rescoping package prepared and submitted to DOE for approval.

Fairbanks Morse Chronology	
September 2003	• DOE approval and resumption of two-cylinder engine testing at Fairbanks Morse.
April 2004	• Successful coal-diesel engine operation on coal water slurry.
July 2004	• Submission of detailed test report on coal-diesel test at Fairbanks Morse. Report included recommended facility modifications to suppress burning particle carryover into the exhaust. Cost proposal prepared for these modifications.
January 2005	• Fairbanks Morse announced restructuring including reduced availability of research engine laboratory.
March 2005	• Search for alternative approaches to completing the demonstration and for sources of matching funds.
October 2005	• Initiation of discussions with DOE on early termination of project due to lack of facility and further cost sharing.
December 2005	• Letter to DOE requesting termination by mutual agreement.

Appendix G. Coal Testing with Various Surfactants

Vendor	Coal Type	Surfactant	Amount Additive	Test Run	Percent of Solids	Results
EERC bb#12 12/4/2003	Sub Bituminous	SLS	2% by weight 437lbs. 1950.8gr.	rack from 10 to 70mm	47.78%	Good results, ran 15minutes coal, 10 minutes diesel No problem during switch over or back.
EERC bb#11 12/5/2003	Sub Bituminous	SLS	2% by Weight 459lbs 1979.5gr.	rack from 10 to 70mm	48.53%	Good results, ran 15 minutes coal, 10 minutes diesel, no problem during switch over or back
CQ 12/9/2003	Bituminous	Triton X	1% by Volume 54 gallons 2080ml	rack from 10 to 70mm	48.93%	Test was mixed, switch over was good, some lumps Injector misfires at only changing rack
CQ 12/9/2003	Bituminous	Triton X	1% by Volume	Rack from 10 to 70mm	47.88%	Retest same barrel as above but lower solids Test worked well ran injector at all rack settings Switched from diesel smoothly, and back
CQ 12/10/2003	Bituminous	Triton X	1% by Volume 54 gallons 2080ml	Rack from 10 to 70mm	48.49%	Poor results after switch over from diesel unable to run above 40mm rack without misfire lower solids did not improve
CQ 1/15/2004	Bituminous	NONE		Rack from 10 to 50mm	48.24%	Ran water only and switched to coal, injector plugged lower solids and repeated test with same results
CQ 1/15/2004	Bituminous	NONE		Rack from 10 to 50mm	47.30%	Ran water only and switch to coal, injector plugged Lower solids Misfire occurrs at any rack setting
CQ 1/16/2004	Bituminous	Triton X	1/2% by volume 50 gallons 500ml	Rack form 10 to50mm	47.09%	testing at 10mm to 30mm rack was ok above 30mm rack causes misfire of injector
CQ 1/16/2004	Bituminous	Triton X	3/4% by volume 420ml	Rack from 10 to70mm	47.32%	Injector testing went well through 70mm rack No misfire occurred during this test switch over good water testing only no diesel
CQ 1/28/2004	Bituminous	Xanthan Gum	115 grams 244lbs. Coal	Rack from 10 to 40mm	47.90%	Testing with this only and would not inject Coal was heavy paste, injector misfire heavy
CQ 1/30/2004	Bituminous	Xanthan Gum	40 grams 0.03%	Rack from 10 to 40mm	47.74%	Tesing with this only, Injector only misfires Coal won't even side out of collector as a lump Paste forms are heavy chuncks.
CQ 2/25/2004	Bituminous	Xanthan Gum	40 grams 1000ml Triton	Rack from 10 to 30mm	48.43%	Testing okay fromwater only till 30mm rack Coal became lumpy and some forming, switched back to water and back to coal, still lumpy Coal is thicker after injection than in the barrel
CQ 2/25/2004	Bituminous	Xanthan Gum Triton X	40 grams 1500ml triton	Rack from 10 to 50mm	48.02%	good injection till 40mm rack, injector doesn't misfire, but the coal becomes semi-solid, paste increase rack doesn't improve preformance Coal is thicker after injection, than in barrel
CQ 2/25/2004	Bituminous	Xanthan gum Triton X	40 grams 2000ml	Rack from 10 to 70mm	48.12%	testing was very good switching from water/coal ran test stand from10 to 70mm for 10 minutes each no misfire, no solids, stayed liquid slurry Coal was thicker after injection than in barrel
CQ 2/26/2004	Bituminous	Dispersant	2420ml 3% by volume	Rack from 10 to 60 mm	47.43%	Testing switching over from water only to coal Solid lumps of paste solutions, no misfire increased rack to higher level didn't show improvement Coal again is thicker after injection than in barrel
CQ 2/27/2004	Bituminous	NONE		Rack from 10 to 70mm	25.47%	very low solids, switched over from water to coal very pasty condition at first and increase rack helped to bring to liquid but still has some lumps. after 50mm rack, injector would misfire sometimes Coal is thicker after injection than in barrel